

Proceedings of:

The Tenth Annual Meeting

The American Society for Precision Engineering

October 15-20, 1995

Stouffer Renaissance Austin Hotel

Austin, Texas

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

*ASPE — The American Society for Precision Engineering
401 Oberlin Road, Suite 108
P.O. Box 10826
Raleigh, NC 27605-0826
Phone (919) 839-8444 Fax (919) 839-8039*

Preface

This book comprises the proceedings of the 1995 Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

Individual readers of this book and nonprofit libraries acting for them are permitted to make fair use of the material contained in this book to be used in teaching or research; provided they copy said paper in total, as written by the author. Permission is granted to quote excerpts from the scientific or technical works in this book with proper acknowledgement of the source to include: the author's name, the book name, and the ASPE tag line. Reproduction of figures and tables is likewise permitted from the contents of this book, provided that the same acknowledgment of the same source and content is printed.

Any other type reproduction, systematic copying or multiple reproduction of any material in this book is expressly prohibited except with the written permission of the ASPE editorial staff, addressed to the Society's headquarters.

ISBN 1-887706-14-3

1995

**American Society for Precision Engineering
401 Oberlin Road, Suite 108
Raleigh, NC 27605-0826**

Printed in the United States of America

Welcome Note

The American Society for Precision Engineering celebrates its Tenth Annual Meeting this year in Austin, Texas. The remarkable growth and success of the Society is reflective of the increasing importance of precision engineering in a wide variety of fields of endeavor, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering problems involving mechanical parts, electronic components, materials and material processing, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

This year's meeting continues the tradition of offering two full days of tutorials presented by some of the foremost precision engineers and scientists in the world. Many participants cite the tutorial program as one of the most valuable features of the meeting. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, material processing, design, precision transducers, surface profilometry, machine tool metrology, and error compensation. The commercial exhibit will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services that are commercially available. The open forum will give attendees the opportunity to express their opinions and pose questions for discussion by the group, enhancing the exchange of information.

The conference organizing committee is proud to present the program for the Tenth Annual Meeting of ASPE. We welcome your participation.

Welcome to Austin.

1995 ASPE Meeting Organizing Committee

*John C. Ziegert, Chairman
University of Florida*

*Thomas G. Bifano, Boston University
Alkan Donmez, Nat'l Institute of Standards & Technology
Thomas A. Dow, North Carolina State University
Anthony E. Gee, Cranfield University
Musa K. Jouaneh, University of Rhode Island
W. Keith Kahl, Oak Ridge National Laboratory
Debra A. Krulwich, Lawrence Livermore Nat'l Laboratory
Don A. Lucca, Oklahoma State University
Frederick G. Parsons, Federal Products Company
Steven D. Phillips, Nat'l Institute of Standards & Technology
Jayaraman Raja, University of North Carolina—Charlotte
Robert G. Wilhelm, University of North Carolina—Charlotte
David H. Youden, Rank Pneumo
Carl A. Zanon, Zygo Corporation*

Program

1995 ASPE Annual Meeting

<i>Time</i>	<i>Sun., Oct. 15</i>	<i>Mon., Oct. 16</i>	<i>Tues., Oct. 17</i>	<i>Wed., Oct. 18</i>	<i>Thurs., Oct. 19</i>	<i>Fri., Oct. 20</i>
— 8:00			Continental Breakfast			
— 9:00	Tutorials	Tutorials	Technical Session 1	Technical Session 4	Technical Session 6	Technical Tours
— 10:00			Break	Break	Break	
— 11:00			Poster Session 1 + Exhibits	Poster Session 2 + Exhibits	Technical Session 7	
— 12:00			Lunch + Committee Meetings	Lunch + Business Meeting	Lunch	
— 1:00						
— 2:00	Tutorials	Tutorials	Technical Session 2	Commercial Session	Technical Session 8	
— 3:00			Break	Break	Break	
— 4:00			Technical Session 3	Technical Session 5	Technical Session 9	
— 5:00						
— 6:00			Hospitality Suite	Keynote Address		
— 7:00		Welcome Reception	Open Forum	Dinner		
— 8:00						
— 9:00				Hospitality Suite		

1995 Annual Meeting

Contents

<i>Preface</i>	<i>ii</i>
<i>Welcome Note—Organizing Committee</i>	<i>iii</i>
<i>Program</i>	<i>iv</i>
<i>Sponsor Members</i>	<i>vi</i>
<i>Cooperating Sponsors—Technical Exhibitors</i>	<i>vii</i>
<i>Lifetime Achievement Awards</i>	<i>viii-ix</i>
<i>Keynote Address/Forum/Commercial</i>	<i>1</i>
<i>Technical Paper Index</i>	<i>2-9</i>
<i>Technical Papers</i>	<i>13-443</i>
<i>Authors Index</i>	<i>445-447</i>
<i>Appendix</i>	<i>449</i>

Sponsor Members of the ASPE

3M Company

3M Center, Bldg. 260-1C-05
St. Paul, MN 55144
(707) 765-3234
Contact: William J. Bryan

ADE Technologies, Inc.

77 Rowe Street
Newton, MA 02166
(617) 969-0600
Contact: John Morgello

Bausch & Lomb

1400 North Goodman Street
Rochester, NY 14692-0450
(716) 338-6754
Contact: Michael J. Clark

Brown & Sharpe

Precision Park
North Kingstown, RI 02852
(401) 886-2358
Contact: David H. Genest

Chardon Tool & Supply Company, Inc.

P.O. Box 291
11993 Ravenna Road #18
Chardon, OH 44024
(216) 286-6440
Contact: Weldon Bennett

Contour Fine Tooling, Inc.

P.O. Box 111, Fisk Mill
7 Water Court
Marlborough, NH 03455
(603) 876-4028
Contact: Jeffrey Tomlin

Cranfield Precision Engineering, Ltd.

Cranfield University
Building 90, Wharley End
Cranfield, Bedford MK43 0AL, England
011-44-1234-752724
Contact: Patrick A. McKeown

Dover Instrument Corporation

P.O. Box 200
200 Flanders Road
Westboro, MA 01581
(508) 366-1456
Contact: John R. Hero

Fabreeka International, Inc.

1023 Turnpike Street
Stoughton, MA 02072
(617) 341-3655
Contact: Robert P. Haley

Lawrence Livermore National Laboratory

P.O. Box 808, L-644
Livermore, CA 94550
(510) 422-1915
Contact: Daniel C. Thompson

Moore Special Tool Company

800 Union Avenue
Bridgeport, CT 06607-0088
(203) 366-3224
Contact: E. Ray McClure

Precitech, Inc.

P.O. Box 409
Keene, NH 03431
(603) 357-2510
Contact: Donald Brehm

Professional Instruments Company

4601 Highway 7
Minneapolis, MN 55416
(612) 927-4494
Contact: Ted J. Arneson, Jr.

Rank Pneumo

Precision Park, P.O. Box 543
Keene, NH 03431
(603) 352-5242
Contact: Leonard E. Chaloux

Tencor Instruments

2400 Charleston Road
Mountain View, CA 94043
(415) 969-6784
Contact: Graham J. Siddall

Tropel Corporation

60 O'Connor Road
Fairport, NY 14450
(716) 388-3469
Contact: Louis Denes

Zygo Corporation

Laurel Brook Road
P.O. Box 448
Middlefield, CT 06455-0448
(203) 347-8506
Contact: Carl A. Zanon

Cooperating Sponsors

Chardon Tool & Supply Co., Inc.

Elsevier Science, Inc.

Photonics Spectra

Technical Exhibitors

ADE Technologies, Inc.

Bal-tec Division

Chardon Tool & Supply Co., Inc.

Contour Fine Tooling, Inc.

Cranfield Precision, Inc.

Digital Instruments, Inc.

Dover Instrument Corporation

Fabreeka International, Inc.

Lion Precision

Moore Tool Company, Inc.

New Focus, Inc.

Phase Shift Technology

Precision Engineering Center—N.C. State University

Precision Engineering Laboratory—University of N.C. at Charlotte

Precision Engineering Research Laboratory—Boston University

Precitech, Inc.

Professional Instruments Co., Inc.

Queensgate Instruments

Rank Pneumo

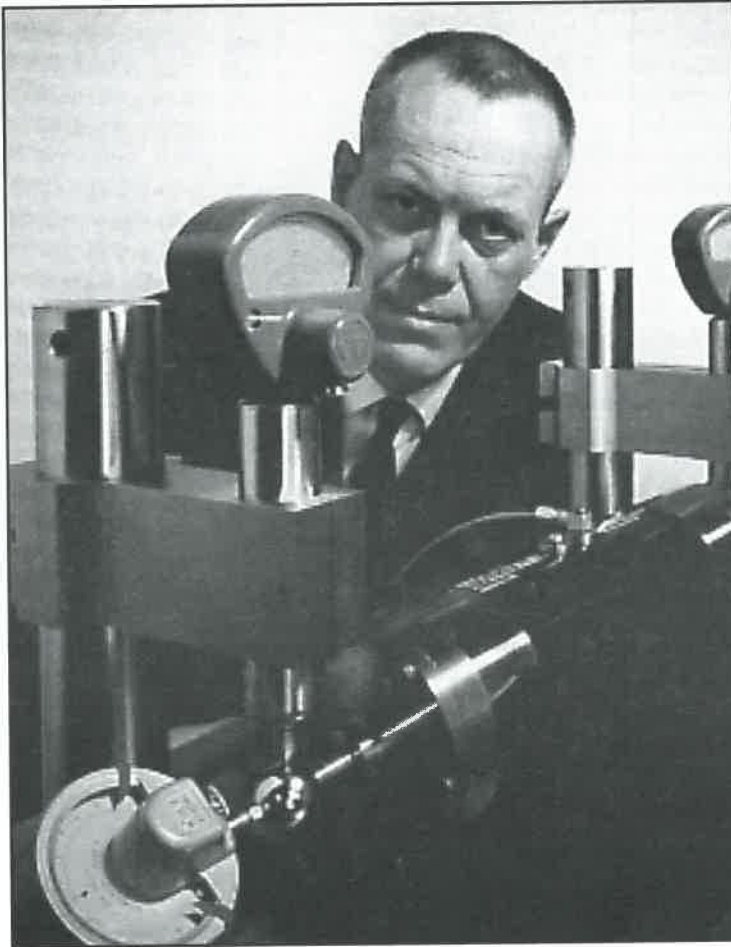
The Timken Company

TROPEL Corporation

Zygo Corporation

1995 Lifetime Achievement Award

Harold E. G. Arneson



Harold was born in Minneapolis in 1925, and studied mechanical engineering at the University of Minnesota. He served in the Air Force during W.W.II and, with his brother Ted, started Professional Instruments Company in 1947. The company entered the ultraprecision engineering field in 1960 by fabricating hollow spherical gyro rotors requiring microinch tolerances. Harold became convinced that hydrostatic air bearings were the optimum route to the accuracies necessary for future requirements in manufacturing and gauging. He invented the dual-hemisphere spindle that became well known in applications at the Oak Ridge National Labs. To overcome the disadvantages of the hemispherical configuration, Harold improved the design and patented the BLOCK-HEAD® spindle in 1969. This spindle became the most widely used ultraprecision air bearing spindle in the world and is standard equipment on Moore diamond turning machines. The BLOCK-HEAD® has seen application in a wide variety of

fields from basic physics research in the value of the gravitational constant to the fabrication of interocular lenses for cataract surgery.

Harold did more than invent air bearing spindles; he contributed to the tool box of precision engineering by contributing ideas and equipment for humidity control, sphere lapping, cylindrical grinding, design of bolted joints, balanced force couplings and bearing error analyzers. Tooling that Harold built was at the heart of many major projects at IBM, Honeywell, 3M, Control Data, Memorex, Bell Labs, Polaroid and others. These tools have served as models for several generations of tool makers and engineers who are continually working to achieve the next decimal place.

Harold died at his retirement home in Fort Myers Beach, Florida in May of 1995. He was a gentleman, a scholar, a scientist, a master craftsman and a great teacher. His life was well spent in the pursuit of knowledge and its practical applications.

1995 Lifetime Achievement Award

Robert R. Donaldson



Dr. Donaldson received his doctorate from the Massachusetts Institute of Technology in 1965. He joined the Mechanical Engineering Department at the University of California at Berkeley in 1964 and taught mechanical design with specialization in bearings and lubrication. In 1966, he began what was to be a long and fruitful association with the University of California's Lawrence Livermore National Laboratory, first as a summer employee, then as a consultant and, finally, in 1970, as a full-time employee. From 1966 to 1974, he contributed original techniques to basic metrology, especially in the area of roundness measurement, and to the design of gages and precision machining systems. In 1978 he became the leader of LLNL's Machine Tool Development Group and, simultaneously, the leader of the Large Optics Diamond Turning Machine (LODTM) project, which was completed in 1983. Bob was responsible for the design and construction of the LODTM and its operation until his retirement from the Lab in 1993. The LODTM is regarded as the world's most precise machine tool and has been used for the fabrication of large, complex, ultraprecise mirrors for applications ranging from space laser weapons to the Keck Telescope. The LODTM design process is regarded as a model for precision machine design and the machine itself incorporates features that are still being assimilated into general practice.

In 1986 Bob was elected to membership in the International Institution for Production Engineering Research (CIRP), one of the most exclusive associations for manufacturing researchers in the world. During his long career, Bob has distinguished himself by his integrity, vision, leadership, and skill as an engineer. As a friend, colleague, and coworker, he has taught many of us how to think about Precision Engineering.

Keynote Address

H. John Wood

Wednesday, October 18, 6:00 p.m.

Dr. H. John Wood is Optics Lead Engineer on the Hubble Space Telescope Project. A graduate of Swarthmore College, Dr. Wood earned an M.A. and Ph.D. in Astronomy from Indiana University. He has been at Goddard Space Flight Center for ten years, where he currently works for the Optics Branch on loan to the HST project. Before joining the Hubble Project, he was Lead Optical Engineer on two recent Goddard projects: for the Mars Observer Laser Altimeter and for the Diffuse Infrared Background Experiment aboard the Cosmic Background Explorer (COBE). Earlier he was assistant to the director at Cerro Tololo Interamerican Observatory (Chile) for two years, and he had served five years as a staff astronomer at the European Southern Observatory in Chile. His career began with six years on the astronomy faculty of the University of Virginia at Charlottesville.

Winner of the 1992 NASA exceptional service medal and the 1994 NASA exceptional achievement medal for his work on COBE and HST, he is the author of 50 research papers in astronomy and space optics. He recently edited special editions of Applied Optics and Optics and Photonics News on the HST first servicing mission. He was co-chair of the HST Independent Optical Review Panel (HIORP) which was charged with the determination of the optical parameters for the HST while on orbit.

Engineering the Correction of the Hubble Space Telescope— The Latest Scientific Results

Engineering the Correction:

- *How the Space Telescope works and how it was built—the flaw in the optics*
- *How the prescription for the HST optics was measured while it was on orbit: star images vs. null corrector metrology*
- *The principles of the optical correction will be illustrated with a short videotape showing a virtual reality prototype of the opto-mechanical systems*
- *The servicing mission STS-61 will be recounted with illustrations*

The New Science Capability:

- *The New Science Capability of HST will be illustrated by examples of objects studied in our solar system, the Milky Way galaxy and its satellites and the most distant and youngest galaxies ever seen*

Open Forum

Tuesday, October 17, 7:00 - 10:00 p.m.

Session Chair: E. Clayton Teague, National Institute of Standards and Technology

The open forum was created to encourage dissemination of new and significant results, as well as observations on precision engineering subjects. All interested conference participants can sign up at the Conference Registration Table for short, informal presentations.

Commercial Session

Wednesday, October 18, 1:30 - 3:00 p.m.

Session Chair: John C. Ziegert

The commercial session will take place on Wednesday afternoon, in a special session where company representatives are invited to make brief presentations on new products associated with precision engineering. Conference participants will have the opportunity to hear about limitations of performance of existing approaches, new approaches relevant to precision engineering, and advances that can be expected in the future.

Technical and Poster Paper Index

The technical program for the 1995 ASPE Annual Meeting contains 110 papers on current research from Asia, Australia, Europe, and South and North America. Authors from national laboratories, universities and industry will share their ideas and conclusions on a variety of topics.

Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster session. This year the poster papers will be close to the exhibits for the duration of the conference, and authors will be present to discuss their work on Tuesday, October 17 and Wednesday, October 18, from 10:30 a.m. to 12:00 p.m.

Once again, in addition to eleven technical sessions, an Open Forum and a Commercial Session will be offered.

Session I Ultraprecision Machining

Tuesday, October 17, 1995 — 8:30 - 10:00 a.m.

Session Chair: Don A. Lucca (Oklahoma State University)

1. **Invited Paper: Semiconductor Industry Precision Engineering Challenges Through 2010***
C. Van Peski (SEMATECH)
2. **Development of a Tool Force/Surface Finish Model for Diamond Turning**
C. Arcona, T. A. Dow (North Carolina State University) page 13
3. **Fabrication of Large Radii Toroidal Surfaces by Single Point Diamond Turning**
J. P. Cunningham, T. A. Marlar, A. C. Miller (Oak Ridge National Laboratory), and R. L. Paterson
(LEXMARK International, Inc.) page 17
4. **Molecular Dynamics Modeling of Thermal Conduction in Microcutting of Metals**
S. Shimada, N. Ikawa, R. Inoue, J. Uchikoshi (Osaka University) page 21

Poster Session I Metrology & Controls

Tuesday, October 17, 1995 — 10:30 a.m. - 12:00 p.m.

METROLOGY

SPM

1. **Vibration Signature Analysis of AFM Images**
G. A. Joshi, J. Fu (National Institute of Standards and Technology), and S. M. Pandit
(Michigan Technological University) page 25

Machine Tools

1. **A Waylube Performance Comparison Using a Telescopic Ball Bar**
T. Ouvreloeil, P. McIntosh (CarboMedics Inc.) page 31
2. **A 5 Degree-of-freedom Capacitive Position Sensor for Straightness Measurements**
J. W. Spronck, M. H. W. Bonse, G. Nijse (Delft University of Technology) page 35

*Abstract not available at press time.

3. **Autonomous Calibration of a Hexapod Machine Tool**
H. Zhuang (Florida Atlantic University) page 39
4. **Using Capacitance Probes to Measure the Limit of Machine Contouring Performance**
D. Martin (Lion Precision) page 43
5. **A Study on Straight Movement Error Detection in the Method of Laser Datum**
Wan D-A., Gao Y., Li Y., Ma S. (Tong-Ji University) page 48

Sensors

1. **Zone-plate Interferometer to Measure the Positioning Error of a Cutting Tool**
T. Nomura, K. Kamiya, H. Miyashiro (Toyama Prefectural University), and H. Tashiro, K. Yoshikawa (Toyama University) page 52
2. **Study of New Type Interferometer of Double Interferogram**
H. Zhu, B. Lu, S. Ye (Tianjin University) page 56
3. **Computer Vision System for 2-D Edge Detection Using Hardware Demodulation**
Jing F., Song J., Li D., Ren Y. (Tianjin University) page 60
4. **A New Fringe-field Capacitive Sensor**
Tian G., Yan B. (Sichuan Union University) page 64
5. **Fibre Optic Sensor on Robot End Effector for Flexible Assembly**
K. L. Yung, W. S. Lau, C. K. Choi, Y. Y. Shan (The Hong Kong Polytechnic University) page 68
6. **Design of a System for Measuring the Power Drawn in a Stirred Tank with Multiple Impellers**
G. Ascanio, B. Castro, A. Martínez, S. Chatwin, E. Galindo (Universidad Nacional Autónoma de México) page 72

Topography

1. **Evaluation of the Dimensions of Sinusoidal Gratings**
T. Ahbe, K. Hasche, K. Thiele (Physikalisch-Technische Bundesanstalt) page 76

CMM

1. **Tracking and Triangulation Approaches to CMM Performance Verification**
P. A. C. Miguel (CT-UNIMEP, Sta. Bárbara d'Oeste), and T. G. King (The University of Birmingham) page 80
2. **A Precision Inspection System for Aircraft Parts Having Thin Features in CAD/CAI Integrated Environment**
H. J. Pahk, W. J. Ahn (Seoul National University) page 84
3. **A New Method of Evaluating the Form Error of Torus**
S. Liu, Y. Li, Z. Zou, J. Li (Tianjin University) page 88
4. **Evaluation of the Position Error of Two-cylinder-workpiece Applying Maximum Material Principle**
S. Liu, J. Li, Z. Zou, Y. Li (Tianjin University) page 92
5. **A Method to Enhance the Quality and Reliability of Inspection Data for Precision Manufacturing Applications**
G. L. Lee, J. Mou (Arizona State University) page 96
6. **Helical Screw Rotor Tolerancing and Coordinate Metrology for Grinding Process Optimization**
I. H. Sutedjo, H. Fannar, J. L. Bowman (Ingersoll-Rand Company), and R. G. Wilhelm (University of North Carolina at Charlotte) page 100
7. **Design and Analysis of a Two-dimensional Aluminum Standard to Calibrate CMMs**
S. Padilla, D. Palomino, G. Ascanio, G. Ruiz, J. Sánchez (Universidad Nacional Autónoma de México) .. page 104
8. **Generalization and Verification of Vectorial Tolerances**
H-T. Yau (National Chung Cheng University) page 108

Surface Finish

1. **Optical Probe for Surface Profiling of Aspherical Mirrors**
P. S. Huang, X. Xu (SUNY Stony Brook) page 112
2. **Noncontact Surface Roughness Measuring System Based on Computer Vision**
You Z., Chen J., Pu X (Tsinghua University) page 116

3. **A Comparative Study of Surface Texture Measurement Using White Light Scanning Interferometry and Contact Stylus Techniques**
B. D. Boudreau, J. Raja, H. Sannareddy (University of North Carolina at Charlotte), and P. J. Caber (WYKO Corporation) page 120
4. **A Study of the Surface Microtopography of Aluminum Alloys in Single Point Diamond Turning**
S. To, W.B. Lee (The Hong Kong Polytechnic University) page 124

CONTROLS

Manufacturing

1. **Feedback Control of a Low-profile Micro-positioning Stage**
P. Ge, M. Jouaneh (University of Rhode Island) page 128

Error Compensation

1. **Error Compensation of Multi-axis Manufacturing Center**
Y. Wang, K. S. Moon, J. W. Sutherland (Michigan Technological University), and G. Zhang, Y. Zang, X. Chu, J. Du (Tianjin University) page 132
2. **A Method to Integrate Neural Networks and Kinematics Based Models for Machine Error Characterization**
H. M. Lee, J. Mou (Arizona State University) page 136
3. **Digital Compensation of Sensors with Frequency Output**
Tian G. (Sichuan Union University) page 140
4. **Computer Modeling of Heterodyne Interferometer Errors**
L. P. Howard, J. A. Stone (National Institute of Standards and Technology) page 143

Machine Tools

1. **Dynamic Tool Path Determination System Based on the Experimentally Obtained Stability Lobe Diagram Using Multi-axis Force**
M. Mitsuishi, T. Ohta, H. Okabe, T. Nagao (The University of Tokyo) page 147
2. **Accuracy Improvement of Machine Tool by Workpiece-referred Control—4th Report: Sensor Position for Plain Turning**
T. Yazawa, Y. Uda, T. Kohno (Tokyo Metropolitan Institute of Technology) page 151
3. **Dynamic Hybrid Position/Force Control of Deburring Robot Manipulators**
R. Venkata Ramana, T. Nagarajan, Y. G. Srinivasa (Indian Institute of Technology) page 155

Session II

Precision Grinding I

Tuesday, October 17, 1995 — 1:30 - 3:00 p.m.

Session Chair: Thomas G. Bifano (Boston University)

1. **Investigation of Acoustic Emission for Use as a Wheel-to-workpiece Proximity Sensor in Fixed-abrasive Grinding**
J. S. Taylor, M. A. Piscotty, K. L. Blaedel, L. F. Weaver (Lawrence Livermore National Laboratory), and D. A. Dornfeld (University of California at Berkeley) page 159
2. **The Dislocation Structure in Alumina Associated with Ultraprecision Grinding**
I. Zarudi, L. C. Zhang (The University of Sydney) page 163
3. **Proposal of High Productivity in Ductile Mode Grinding of Brittle Materials**
A. Kanai, M. Miyashita, M. Sato (Ashikaga Institute of Technology), and M. Daito (Nissin Machine Works Ltd.) page 167
4. **Shear-mode Grinding Force Criteria in Grinding with Serrate Wheels**
H. Hashimoto, K. Imai (Kanagawa Institute of Technology) page 171
5. **High-efficiency Mirror Finishing of Ceramics with Diamond Wheel**
X-J. Wang, K. Syoji, T. Kuriyagawa (Tohoku University) page 175

Session III

Precision Grinding II

Tuesday, October 17, 1995 — 3:30 - 5:00 p.m.

Session Chair: W. Keith Kahl (Oak Ridge National Laboratory)

1. **Glass Polishing for Optical Communication Devices—Characterization of Slightly Damaged Surface**
S. Matsui, H. Ohira, K. Matsunaga, Y. Kikuya (NTT Interdisciplinary Research Labs.) page 179
2. **Grinding of Micro Aspherical Surface**
H. Suzuki, S. Kodera, S. Hara (Mitsubishi Electric Corporation), and K. Tanaka, H. Takeda (Toshiba Machine Co., Ltd.) page 183
3. **Time Dependent Voltage & Current Behavior for Electrolytic In-process Dressing of Bound Abrasive Wheels (ELID)**
K. D. Growsky, D. E. Johnson, F. Demarest (Zygo Corporation) page 187
4. **Electrolytic Dressing of Bronze Bonded Diamond Grinding Wheels**
H. Caggiano, T. G. Bifano (Boston University) page 191
5. **In-process Dressing of Grinding Wheels for Brittle Materials**
K. W. Sharp, R. O. Scattergood (North Carolina State University) page 195

Session IV

Micro-positioning

Wednesday, October 18, 1995 — 8:30 - 10:00 a.m.

Session Chair: Musa K. Jouaneh (University of Rhode Island)

1. **Spring-dominated Regime Design of a High Load Capacity, Electromagnetically Driven X-Y- θ Stage**
K-S. Chen, D. L. Trumper, M. Williams (Massachusetts Institute of Technology), A. Monterio (University of Warwick), and S. T. Smith (University of North Carolina at Charlotte) page 199
2. **A Versatile XY Stage with a Flexural Six-degree-of-freedom Fine Positioner**
J. A. Gargas, R. K. Dorval, D. Mansur, H. D. Tran, D. Carlson, M. Hercher (OPTRA, Inc.) page 203
3. **Long Range, Direct Drive XY-($\Phi=0$)-Stage with Sub-micrometer Resolution**
G. M. Bonnema, F. Auer, J. de Lange, H. F. van Beek (Delft University of Technology) page 207
4. **A New Hybrid Concept for a Long Stroke Fast-tool-servo System**
M. Weck, H. Oezmeral, K. Mehlkopp, T. Terwei (Fraunhofer-Institut für Produktionstechnologie, IPT) page 211
5. **A New Inchworm® Motor for Industrial Nanopositioning Applications**
D. Henderson (Burleigh Instruments, Inc.) page 215

Poster Session II

Processing & Design

Wednesday, October 18, 1995 — 10:30 a.m. - 12:00 p.m.

PROCESSING

Turning

1. **Diagnostic of Diamond Turning Machining through FFT Analysis of Finished Surfaces**
A. J. V. Porto, J. G. Duduch, L. C. R. Carpinetti, R. G. Jasinevicius, R. Tsunaki (Universidade de São Paulo), and A. E. Gee (Cranfield University) page 219
2. **Temperature Modeling for Orthogonal Diamond Machining**
B. C. Rao, R. X. Gao, C. R. Friedrich (Louisiana Tech University) page 223

3. **Relationship Between Catastrophic Fracture and Electric Resistance of Cutting Tools—
A Trial for Catastrophic Tool Fracture Prediction**
K. Uehara (Toyo University), and N. Obitsu (Kiri Machine Mfg. Co., Ltd.) page 227
4. **Inverse Technique for Analyzing the Orthogonal Cutting Process**
W. Xu, J. Genin, Q. Dong (New Mexico State University) page 231
5. **Chip Segmentation in Machining AISI 52100 Steel**
M. A. Davies, C. J. Evans, K. K. Harper (National Institute of Standards and Technology) page 235

Grinding

1. **Grinding Mechanisms of Resinoid Bonded Diamond Wheels with Ultra Fine Grains**
J. Tang, D. Dornfeld (University of California at Berkeley), and K. Syoji, T. Kuriyagawa
(Tohoku University) page 239
2. **Part Speed Effects on Surface Roughness in Precision Grinding**
G. E. Storz II, T. A. Dow (North Carolina State University) page 243
3. **Grinding Temperature in Ceramic Workpieces**
S-X. Liang, O. F. Devereux, Y.Y. Li (University of Connecticut) page 247
4. **Implementation of a Real-time Grinding Process Estimator**
H. E. Jenkins, T. R. Kurfess (Georgia Institute of Technology) page 251
5. **In Process Monitoring and Control of Internal Grinding in High Strength Steel**
P. Ma, M. Wang (Nanjing University of Aeronautics & Astronautics) page 255
6. **Framework for Assessing Key Variable Dependencies in Loose-abrasive Grinding and Polishing**
J. S. Taylor, D. M. Aikens, N. J. Brown (Lawrence Livermore National Laboratory) page 260

Polishing

1. **N/C Polishing Program: Continued Force Wear and Part Correction Experiments**
P. R. Hannah, R. D. Day, D. J. Hatch (Los Alamos National Laboratory), and E. R. McClure
(Moore Tool Co.) page 264
2. **A Computer-controlled Polishing Machine for Strongly Aspheric Optics**
Q. Kong, R. Zimmermann, M. High (Applied Physics Specialties Ltd.) page 268

Modeling

1. **Determinism in Dice Throwing and the Transition to Chaos**
L. Hale (Massachusetts Institute of Technology) page 272
2. **Analytical Determination of the Temperature Distribution Within a Hollow Hemisphere
During a Machining Process**
J. C. Shipley (Los Alamos National Laboratory), and G. P. Mulholland (New Mexico State University) ... page 276

Other Novel Processing

1. **Thrust Force and Wall Roughness in Microdrilling**
C. Friedrich, Y. She, R. Muthu, C. Dave (Louisiana Tech University) page 280
2. **Mechanical Micromilling of PMMA**
C. Friedrich, B. Kikkeri, T. Nagarajan (Louisiana Tech University) page 284
3. **A Preliminary Study on the Effect of Tool Flank Wear on Surface Finish in End Milling**
S. N. Melkote (Georgia Institute of Technology) page 288
4. **Experimental Investigations Into Electro-chemical Spark Machining of Kevlar Epoxy Composites**
V. K. Jain, S. K. Choudhury, V. Nesarikar (Indian Institute of Technology) page 292
5. **Cold Form Tapping of Internal Threads for Landing Craft in High Strength Steel**
P. Ma, M. Wang (Nanjing University of Aeronautics & Astronautics) page 296

DESIGN

Precision Machining

1. **Design of a Long Range Fast Tool Servo System Using Magnetic Servo Levitation**
B. A. Stancil, H. M. Gutierrez, P. I. Ro (North Carolina State University) page 301

2. **A High Precision Five-degree-of-freedom Micropositioner Based on Electromagnetic Driving Principle**
W. Wang, T. He (Louisiana Tech University), and I. Busch-Vishniac (The University of Texas at Austin) . . page 305
3. **Influence of Form Deviations on the Dynamic Performance of Microhydraulic Actuators**
K. Sadashivappa, M. Singaperumal, K. Narayanasamy (Indian Institute of Technology) page 309
4. **Development of a Parallel Architecture Five-axis Coordinate Measuring Machine**
B. Jokiel, Jr., J. C. Ziegert (University of Florida) page 313
5. **Micro-impulse Generation Digging Tool for Micro Surgery**
M. Nakao, Y. Hatamura, R. Asai, T. Yano (The University of Tokyo) page 317
6. **Design of Spur and Helical Gears***
J. Mandák (Language Studies Institute MTA). page 321
7. **A New Method for Improving Dynamic Stiffness of Advanced Ceramic Structures**
P. Scagnetti (Wilbanks International), E. Marsh (The Pennsylvania State University), and
A. H. Slocum, S. Nayfeh (Massachusetts Institute of Technology) page 322
8. **An Experimental Study on the High-precision Linear Motion Table with Magnetic Bearing Suspension**
Y. Joung, S-C. Jung, D-C. Han (Seoul National University), and I-B. Chang
(Kangwon National University) page 328
9. **Analytical and Experimental Identification of Nonlinearities in a Single Nut, Preloaded, Precision Ball Screw**
J. F. Cuttino (The University of Alabama), and T. A. Dow (North Carolina State University) page 332

Bearings

1. **Analytical Expression of Damping Ratio of Air-lubricated Slideways for Ultraprecision Machine Tools**
S. Kato (Nippon Institute of Technology) page 336
2. **Modeling of Self-compensated Bearings for High Stiffness**
B. Zhang, Y. Sun (University of Connecticut) page 340

Machine Tools

1. **Design of Dynamically Stiff Precision Machines for Low Damage Grinding of Brittle Materials. Tetraform™ a New Approach***
P. Morantz, P. Shore (Cranfield Precision Engineering Ltd.) page 344
2. **A Compliance Model of the Roller Contact Interface for a Friction Drive Used on Ultraprecision Machine Tools***
B. N. Damazo (National Institute of Standards and Technology), A. E. Gee (Cranfield University),
and A. H. Slocum (Massachusetts Institute of Technology) page 345
3. **Application of Linear Scale With 1 nm Resolution to Ultraprecision Machine Tools**
A. Kanai, M. Miyashita, T. Kitahara (Ashikaga Institute of Technology) page 346

Session V

Precision Transducers & Measurement

Wednesday, October 18, 1995 — 3:30 - 5:00 p.m.

Session Chair: Anthony E. Gee (Cranfield University)

1. **A New Probe for Friction Measurements at Very Small Scales**
L. A. J. Davis, D. K. Bowen, D. G. Chetwynd (University of Warwick) page 350
2. **A Two-degrees-of-freedom Capacitive Position Sensor With Improved Measurement Algorithms**
M. H. W. Bonse, J. W. Spronck (Delft University of Technology) page 354

* Extended abstract not available at press time.

3. **Extending the Accuracy and Precision of In-situ Ultrasonic Thickness Measurements**
D. E. Dunn, J. R. Cerino (Hughes Danbury Optical Systems) page 358
4. **Evaluation of High Precision Triangulation Sensors for Coordinate Measurement**
M. V. Ananda, J. A. Jalkio, L. Konicek (CyberOptics Corporation) page 364
5. **Measurement of Friction at Light Loads in Polypyrrole Thin Film Bearings**
X. Liu, D. G. Chetwynd, J. W. Gardner (University of Warwick), S. T. Smith (University of North Carolina at Charlotte), and C. Beriet, P. N. Bartlett (University of Southampton) page 368

Session VI

Micro Devices

Thursday, October 19, 1995 — 8:30 - 10:00 a.m.

Session Chair: David H. Youden (Rank Pneumo, Inc.)

1. **Invited Paper: An Architecture for Agile Assembly**
R. L. Hollis, A. Quaid (Carnegie Mellon University) page 372
2. **Breakthroughs in 3-D Ultra Fine Shape Fabrication by Dynamic FAB Etching**
M. Nakao, S. Tanaka, Y. Hatamura (The University of Tokyo), and M. Hatakeyama, K. Ichiki (Ebara Research Co., Ltd.) page 376
3. **Design and Fabrication of a Two Dimensional Valveless Micropump**
W. K. Kahl, C. M. Egert, K. W. Hylton (Oak Ridge National Laboratory) page 380
4. **Micro-pattern Control by Miniature Robots in Vapor Deposition Process**
H. Aoyama, K. Mizutani, F. Iwata, A. Sasaki (Shizuoka University) page 384

Session VII

Surface Profilometry

Thursday, October 19, 1995 — 10:30 a.m. - 12:00 p.m.

Session Chair: Jay Raja (University of North Carolina at Charlotte)

1. **New Algorithms for the Evaluation of Discrete Point Measurement Data and for Sample Point Selection on Surfaces with Systematic Form Deviations**
K. D. Summerhays, R. P. Henke, R. M. Cassou (University of San Francisco), J. M. Baldwin (Sandia National Laboratories), and C. W. Brown (Allied Signal Aerospace) page 388
2. **Interferometric Calibration of a Capacitance Gauge for AFM Step Height Metrology***
V. W. Tsai, E. D. Williams (University of Maryland at College Park), and J. Schneir, T. McWaid, R. Dixson (National Institute of Standards and Technology)
3. **Recent Advances in Stylus Based 3-D Surface Measurement and Characterization**
P. J. Scott (The University of Birmingham), and E. Morrison (Rank Taylor Hobson Ltd.) page 392
4. **Efficient Characterization of Surface Topography in Cylinder Bores**
P. Pawlus (Rzeszow University of Technology), and D. G. Chetwynd (University of Warwick) page 396
5. **Application of Confocal Scanning Optical Microscope to Three Dimensional Surface and Form Error Characterization**
G. Udupa, M. Singaperumal, R. S. Sirohi, M. P. Kothiyal (Indian Institute of Technology) page 400

*Abstract not available at press time.

Session VIII

Machine Tool Metrology

Thursday, October 19, 1995 — 1:30 - 3:00 p.m.

Session Chair: Robert G. Wilhelm (University of North Carolina at Charlotte)

1. **Modeling and Prediction of Thermally Induced Error Maps in Machine Tools Using a Laser Ball Bar and a Neural Network**
N. Srinivasa (University of Illinois at Urbana-Champaign), and J. C. Ziegert (University of Florida) page 404
2. **Rapid Mapping of Volumetric Errors**
D. Krulewich, L. Hale, D. Yordy (Lawrence Livermore National Laboratory) page 408
3. **Numerical and Experimental Study of Ball-bar Dynamics During Probing**
Y-Z. Dai (Brown & Sharpe Manufacturing Company) page 412
4. **Automatic Measurement of Axial and Tilt Error Motions of an Aerostatic Spindle Using Optical Interferometry**
A. Idowu, A. E. Gee (Cranfield University) page 416
5. **On-machine Acceptance of Machined Components**
A. Hazelton (Sandia National Laboratories) page 420

Session IX

Error Compensation

Thursday, October 19, 1995 — 3:30 - 5:00 p.m.

Session Chair: Debra Krulewich (Lawrence Livermore National Laboratory)

1. **Correlation Models for Improved Machine Tool Accuracy**
N. Srinivasan, F. Farabaugh, R. G. Wilhelm, R. J. Hocken (University of North Carolina at Charlotte). page 424
2. **Error Compensation for CMM Touch Trigger Probes**
W. T. Estler, S. D. Phillips, B. Borchardt, T. Hopp, C. Witzgall, M. Levenson, K. Eberhardt, M. McClain (National Institute of Standards and Technology) and Y. Shen, X. Zhang (George Washington University) page 428
3. **Accuracy Enhancement in Profile Grinding Through Utilization of Post Process Inspection with Feedback**
T. M. Dalrymple, J. C. Ziegert (University of Florida) page 432
4. **Active Thermal Deformation Compensation Based on Internal Monitoring Using a Neural Network with the Genetic Algorithm Method**
M. Mitsuishi, T. Okumura, N. Sugita, T. Nagao, Y. Hatamura (The University of Tokyo) page 436
5. **Compensation of Errors Detected by Process-intermittent Gauging**
H. T. Bandy, D. E. Gilsinn (National Institute of Standards and Technology) page 440

Technical Papers



A S P E

DEVELOPMENT OF A TOOL FORCE/SURFACE FINISH MODEL FOR DIAMOND TURNING

Chris Arcona, Graduate Student
Thomas A. Dow, Professor
Precision Engineering Center
North Carolina State University
Raleigh, NC 27695-7918

INTRODUCTION

The components of the steady-state cutting forces in diamond turning contain a wealth of information about the state of the material removal process. Changes in these forces should indicate a change in both the geometry of the tool and the finish of the machined surface. For this reason, it is very important to understand and model the relationships between machining conditions and tool forces. It would be ideal if this model could be based on standard material parameters such as hardness, yield strength and modulus of elasticity as well as the geometry of the tool. Unfortunately, such a model is not currently available for diamond turning, although considerable effort has been expended in an attempt to explain the cutting process. One limit to the development of this model is a lack of detailed experimental data to corroborate the analysis. A new technique to overcome this limitation has been developed. This technique involves machining a split workpiece which can be separated to view the edge of the chip [1]. The method produces high resolution images of the material flow that forms the chip and can be used to corroborate the theoretical models of the force components.

CHIP GEOMETRY

Experiments to produce chip cross sections have been conducted with plated surfaces of copper and electroless nickel. A tool with a 5.08 mm nose radius, 0° rake angle and 8° clearance angle was used to make the cuts. The tool was worn by cutting 6061-T6 and chip

cross sections were made at three stages of tool wear (50, 180 and 300 nm edge radii; 0, 1.4 and 2.4 μm flank wear lands, respectively). These tool dimensions were found using a scanning electron microscopy technique developed by Drescher [2]. The components of the tool force are monitored with a three axis load cell (Kistler 9251A) and collected with a MASSCOMP data acquisition system.

Tool Forces. A typical graph of tool force vs. time is shown in Figure 1.

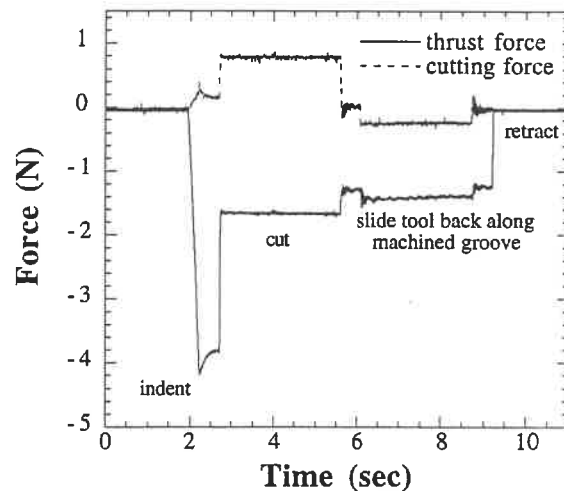


Figure 1. Tool force during slow speed cut in plated copper.

This output corresponds to cutting plated copper at a 1.68 μm depth of cut with the tool having a 0.18 μm edge radius and 1.4 μm wear land. The cutting speed was 25 mm/min and the sampling frequency was 500 Hz. Each step of the cutting process (tool indentation, cutting, sliding back along the machined groove and tool retraction) is labeled in Figure 1. The

average cutting force and thrust force were measured as 0.81 N and 1.62 N, respectively. The redundant forces, or the forces not associated with material removal, are measured by sliding the tool back along the machined surface. These forces were 1.37 N (thrust) and 0.21 N (cutting). The ratio of these forces is 0.15 and yields an estimate for the coefficient of friction between the tool and the workpiece. The constant force levels in Figure 1 reflect a uniform depth of cut.

Chip Cross Sections. A cross section of a plated copper chip produced with the sharp tool (50 nm edge radius) at a cutting speed of 25 mm/min is shown in Figure 2. The depth of cut reflected in the micrograph is 3.9 μm . In Figure 2, note that narrow zones of shear originate at the location of the tool tip and evenly separate the chip into segments or lamella. The average thickness of the lamella in Figure 2 is 0.7 μm .

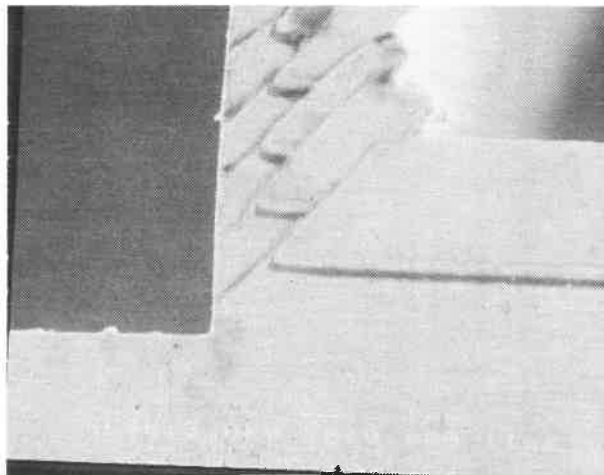


Figure 2. Plated copper chip, 25 mm/min, sharp tool; 7,000X.

The segmented scratch parallel to the workpiece surface shows that the deformation within each segment is very small compared to that occurring in the shear zone. The lamella thickness has been observed to vary linearly with the corresponding depth of cut, as other researchers have found [3]. Figure 2 along with five other chip cross sections at depths of

cut between 2 and 10 μm showed that the sheared regions separating the lamella are straight, very thin and that they are inclined 45° to 50° from horizontal.

The experimental technique revealed differences in chip structure due to tool geometry. Figure 3 shows another cross section of a plated copper chip. This chip was also produced at a speed of 25 mm/min, but with the worn tool (300 nm edge radius and 2.4 μm wear land). The depth of cut at this section is about 2.2 μm . The shear angle in Figure 3 varies between 36° and 40° (about 25% less than in the sharp tool case) while the lamella thickness is about 0.9 μm . The rounded region at the chip root in Figure 3 also suggests that elastic deformation plays a significant role in cutting with the dull tool.

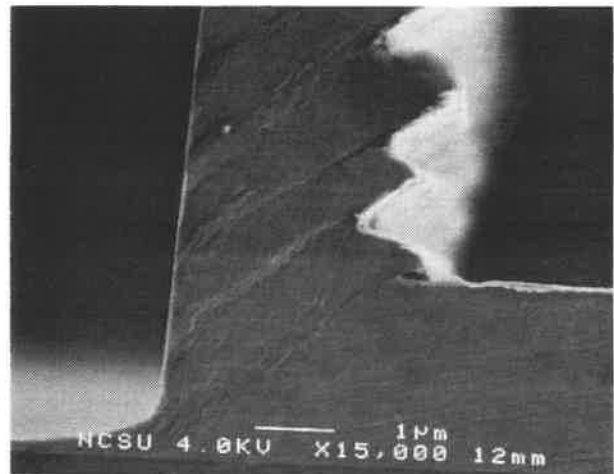


Figure 3. Plated copper chip, 25 mm/min, dull tool; 13,500X.

Cross sections of electroless nickel chips also show material and tool dependence of the chip structure. Figures 4 and 5 show cross sections of electroless nickel chips machined at 25 mm/min. The depth of cut in both micrographs is 1.5 μm . The chip in Figure 4 was created with the sharp tool while that in Figure 5 is the result of cutting with the tool of 300 nm edge radius and 2.4 μm wear land. As for plated copper, the chips are separated into lamella by thin, straight zones of shear. Differences in the shear angle and lamella thickness are also

observed for variations in cutting conditions. For chips machined with the sharp tool, the shear angle is about 22° . Machining electroless nickel with the worn tool resulted in shear angles of about 16° , as demonstrated by Figure 5. This reduction in shear angle is about 25%, as observed in plated copper chips. Furthermore, the lamella thickness is $0.9 \mu\text{m}$ in Figure 4 but increases to $1.4 \mu\text{m}$ as the tool edge radius increases, as shown by Figure 5.

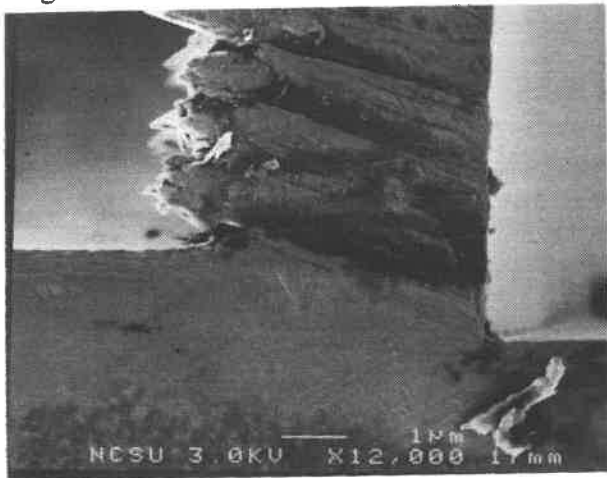


Figure 4. Electroless nickel chip, 25 mm/min, sharp tool; 8,400X.

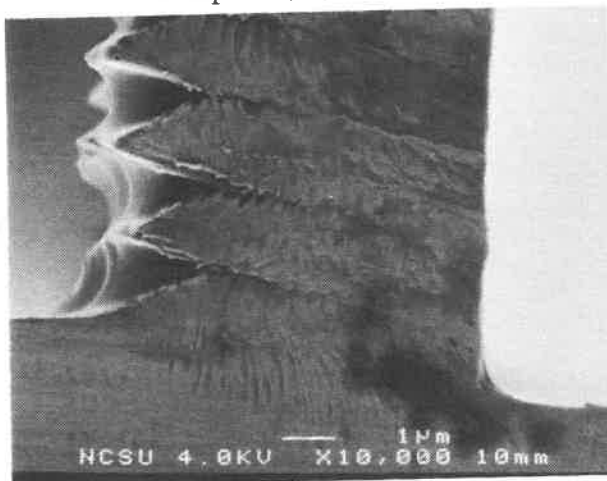


Figure 5. Electroless nickel chip, 25 mm/min, worn tool; 7,000X.

FORCE MODEL

An analysis of the chip generation shown in Figures 2 - 5 can form the basis for a tool force model. A free body diagram of a chip segment in the process of being formed appears in Figure 6. Chip formation consists of

deformation in the region of the segment adjacent to the tool rake face followed by shear in a thin region inclined at some angle, ϕ , to horizontal.

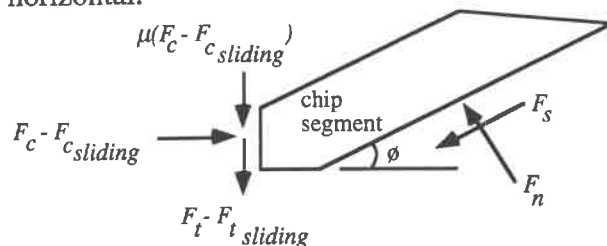


Figure 6. Forces on a chip segment.

The forces $F_t - F_{tsliding}$ and $F_c - F_{csliding}$ are the cutting and thrust forces due only to material removal while $\mu(F_c - F_{csliding})$ is a frictional force between the tool and the chip. F_s and F_n are the shear and normal forces on the shear zone, respectively. It is assumed that there is negligible shear force between the workpiece and the segment previously formed. An additional friction force parallel to the motion of the tool, $\mu(F_t - F_{tsliding})$, acts on the flank region of the tool so that the equation for the total cutting force, F_c , is

$$F_c = F_s \cos \phi + F_n \sin \phi + \mu F_t \quad (1)$$

The equation for the total thrust force, F_t , is

$$F_t = F_n \cos \phi - F_s \sin \phi + F_{tsliding} + \mu(F_c - F_{csliding}) \quad (2)$$

Thus, the force model requires knowledge of the shear and normal forces acting on the shear zone and a model for the redundant thrust force due to the sliding of the tool along the machined surface. The redundant cutting force, $F_{csliding}$, can be calculated once an estimate for the coefficient of friction is obtained.

Equations (1) and (2) have been used to estimate the forces obtained in machining electroless nickel. In performing these calculations, the material is regarded as perfectly plastic. The normal stress is taken as

the uni-axial yield stress or one-third the measured Vickers hardness of 5.25 GPa. From a von Mises yield criterion, the shear stress is $1/\sqrt{3}$ times the yield stress. The coefficient of sliding friction at the tool/workpiece interface is calculated to be 0.2 from force graphs such as that in Figure 1.

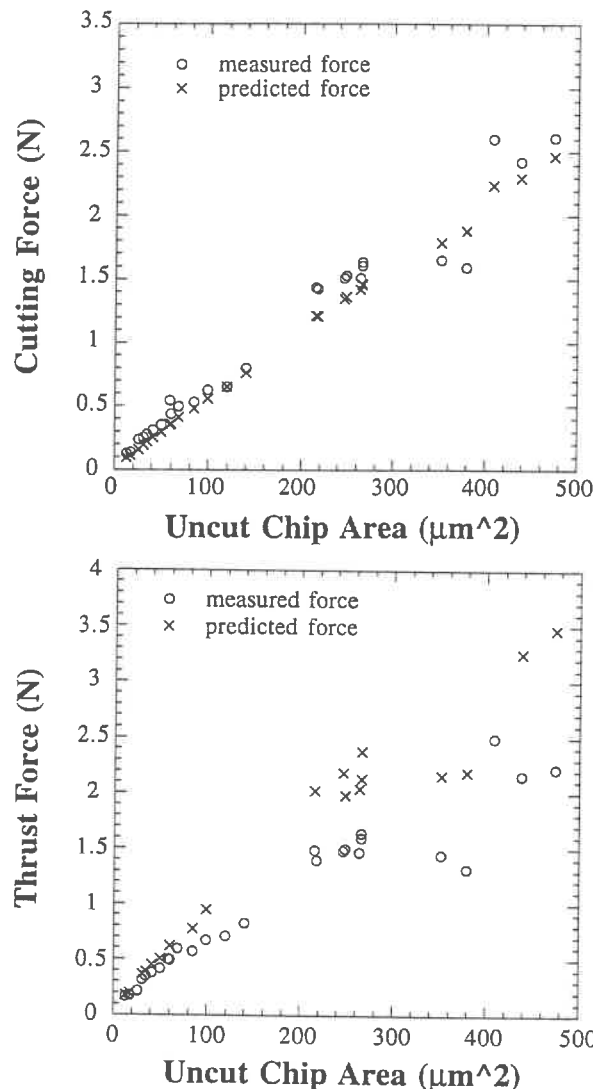


Figure 7. Measured and predicted tool forces for electroless nickel cut with a sharp tool.

The graphs of tool force vs. undeformed chip area for electroless nickel cut with the sharp tool are shown in Figure 7. The shear angle was obtained from micrographs of chip cross sections (see Figure 4). Also, since no model

for the redundant force $F_{t \text{ sliding}}$ has been determined, those values measured during the cutting tests were used in Equation (2). Figure 7 shows the good correlation between the measured and predicted forces for electroless nickel. For the plated copper, the cutting forces are underestimated by about 25%. This result could be due to strain hardening effects during machining which would act to increase the shear strength of the material.

SUMMARY

A technique to produce cross sections of diamond turned chips has been developed. Chips have been formed in plated surfaces of electroless nickel and copper with sharp and dull tools. Investigation of these chips in the SEM has shown that chip formation is a process involving plastic deformation of the workpiece in the vicinity of the tool edge followed by shear in a narrow zone. This process results in relatively undeformed segments separated by thin intense zones of shear inclined at some angle to horizontal. The relationship between this shear angle, material properties and cutting parameters is being studied. A force model based on the observed material flow has yielded good correlation between measured and predicted forces for slow speed non overlapping cuts in electroless nickel and plated copper.

REFERENCES

1. Arcona, C. and Dow, T.A. "A New Technique for Studying the Chip Formation Process in Diamond Turning." *Precision Engineering*, to be published in Fall 1995.
2. Drescher, J.D. "Scanning Electron Microscope Technique for Imaging a Diamond Tool Edge." *Precision Engineering*, v. 15, Jan. 1993, pp. 112 - 114.
3. Ueda, K. and Manabe, K. "Chip Formation Mechanism in Microcutting of an Amorphous Metal." *Annals of the CIRP*, v. 41, no. 1, 1992, pp. 129 - 132.

Fabrication of Large Radii Toroidal Surfaces by Single Point Diamond Turning

Joseph P. Cunningham, Troy A. Marlar, Arthur C. Miller
Oak Ridge National Laboratory, Oak Ridge, TN.
Robert L. Paterson
LEXMARK International, Inc., Lexington, KY.

Introduction

An unconventional machining technique has been developed for producing relatively large radii quasi-toroidal surfaces which could not normally be produced by conventional diamond turning technology. The maximum radial swing capacity of a diamond turning lathe is the limiting factor for the rotational radius of any toroid. A typical diamond turned toroidal surface is produced when a part is rotated about the spindle axis while the diamond tool contours the surface with any curved path. Toric surfaces sliced horizontally, have been used in laser resonator cavities. Reference [1] describes the metrology for one such surface. This paper will address the fabrication of a special case of toroids where a rotating tool path is a circle whose center is offset from the rotational axis of the toroid by a distance greater than the minor radius of the tool path. (See Figure 1.) The quasi-toroidal surfaces produced by this technique approximate all asymmetrical combinations of concave/convex sections of a torus. Other machine configurations have been reported which offer alternative approaches to the fabrication of concave asymmetric aspheric surfaces. [2]

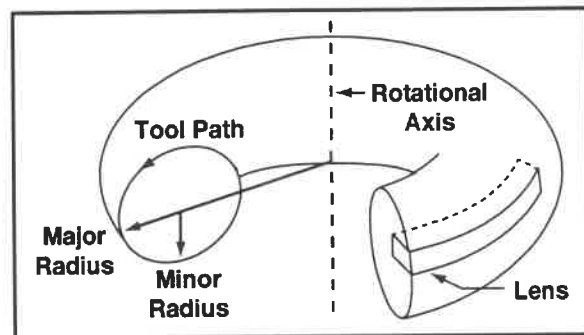


Figure 1. Torus Geometry

Prototypes of unique lenses each having two quasi-toroidal surfaces were fabricated in the Ultraprecision Manufacturing Technology Center at Oak Ridge National Laboratory. These lenses form key components of a scanned laser focusing system. (see Figure 2.) As an example of the problem faced, the specifications for one of the surfaces was equivalent to a section of a torus with a two meter diameter hole. The lenses were fabricated on a Nanoform 600 diamond turning lathe made by Rank Taylor Hobson Inc. in Keene, New Hampshire. This is a numerically controlled two axis T-base lathe with an air bearing spindle and oil hydrostatic slides. The maximum radial swing for this machine is approximately 0.3 meters.

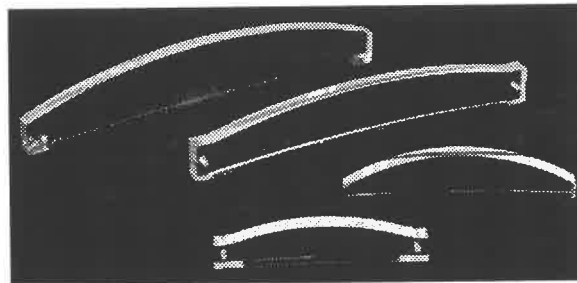


Figure 2. Laser Scanning Lenses

Description of Technique

The diamond turning lathe is used in a fly-cutting mode by rotating a hollow tool-holding fixture about the spindle axis. (See Figure 3.) The work piece is held stationary with respect to the Z slide. The X and Z slides are programmed to contour the major radius of the toroid while the diamond tool rotates with the spindle and cuts at a predetermined fixed minor radius. This arrangement produces a quasi-toroidal surface which deviates slightly from the desired toroidal surface. An elliptical error results when the cutting plane of the tool is not perpendicular to a tangent of the major circumference at every point along the curve. The magnitude of the error is a

function of the angle between the cutting plane of the tool and a line normal to the surface of the major radius. For a given quasi-toroidal surface, the figure error of the surface decreases as the height off axis decreases and the angle approaches 0 degrees. (See Figures 4. and 5.)

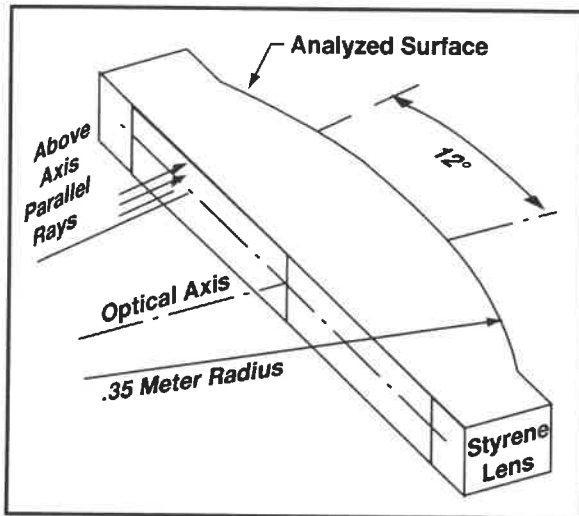


Figure 4. Error Analysis Geometry

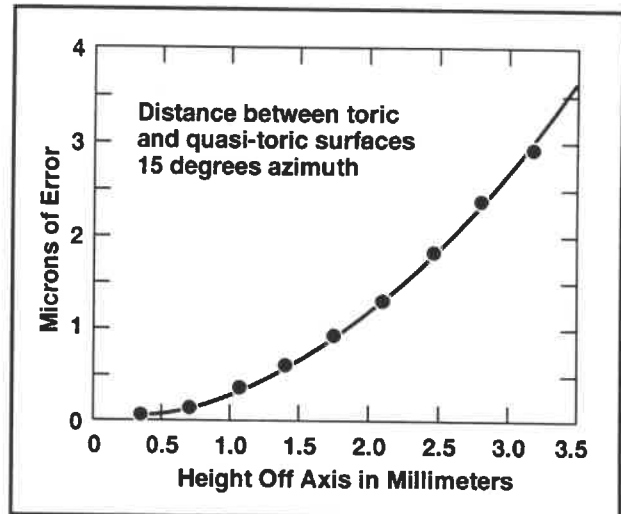


Figure 5. Error as a Function of Height

Fixtures

Figure 3. illustrates the two basic fixtures used with the lathe to machine the quasi-toroidal optical surfaces. One fixture provides stationary support of the lens blank relative to the Z axis while the other holds the diamond tool and rotates with the air bearing spindle. The spindle and the rotating pipe shaped fixture are carried on the X axis of the diamond turning lathe. The lens holding fixture consists of a large solid block with a cantilevered support bar to which the lens is clamped. The cross-section and stiffness of this steel bar was maximized based on the clearance required for the tool path of the minor radius of the toroid. The cantilevered support is required to provide clearance inside the tool path because the lens actually transverses inside the rotating fixture during the cut. This setup allows for a convex surface to be produced for the minor radius while the X and Z slides are programmed to simulate the concave or convex major radius of the toroidal surface. The design of the rotating fixture allows for the tool holder to be mounted on the inside or outside diameter of the pipe. This provides a means of cutting large diameter concave surfaces. A second rotating tool holder consisting of a round solid bar with the diamond tool mounted on its outside diameter is used to cut concave surfaces that have small minor radii. (See Figures 6. and 7.)

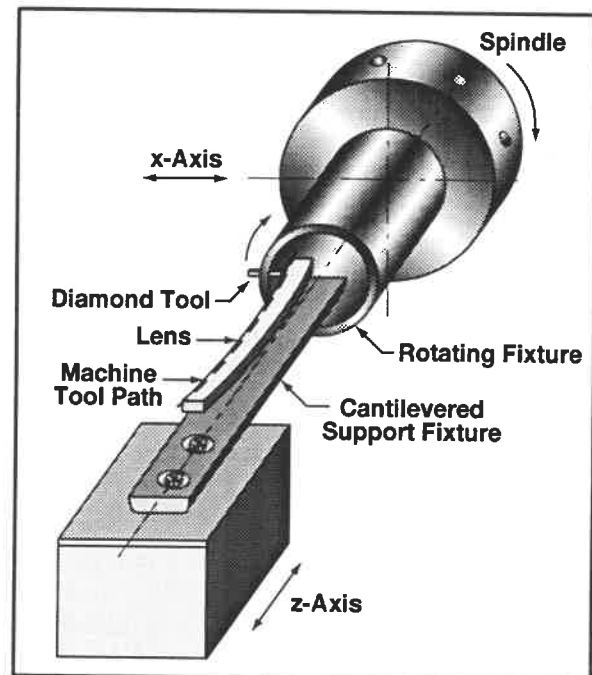


Figure 3. Toroid Fabrication Setup

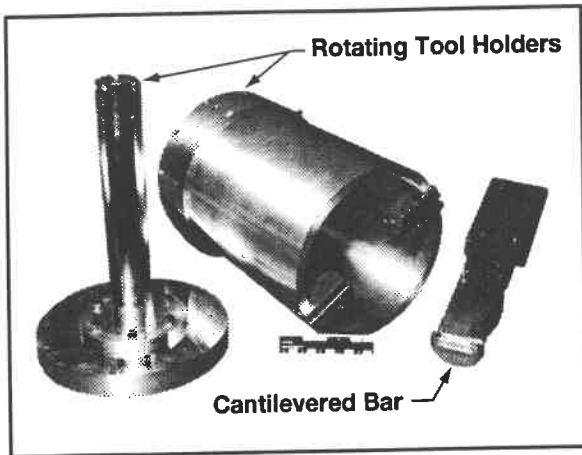


Figure 6. Fixtures

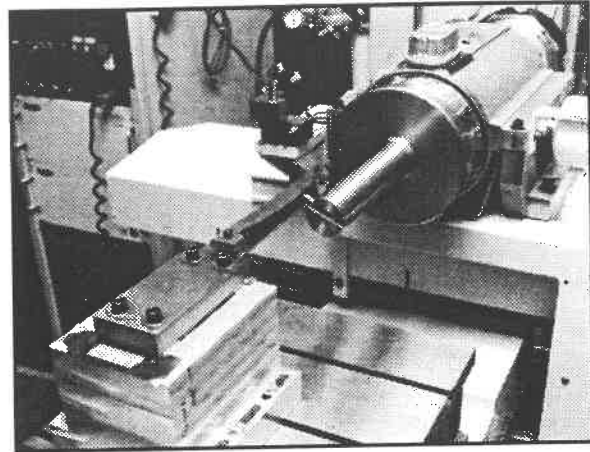


Figure 7. Setup for Concave Minor Radius and Convex Major Radius

Lens Blank Preparation

The lens blanks were fabricated on a CNC mill from optical quality cast REXOLITE® 1422 polystyrene blocks. [3] Dowel pin holes were precisely located at each end of the blanks. The toroidal surfaces were rough machined as cylindrical surfaces on the sides of the blanks perpendicular to their optical axis. After rough machining, the blanks were tested with a polarizer and those having minimum stress and birefringence were selected. Next, the blanks were mounted on a flat disk fixture using dowel pins and thin double sided tape. The dowel pins prevented the blanks from creeping on the double sided tape by providing extra stability in the radial direction against centrifugal force. Each blank was diamond turned smooth and flat on both sides to a precise thickness. Rubber tipped set screws located in the fixture under each lens were used to push the lens free of the double sided tape without producing excessive stress.

Tool Path Radius

The technique for accurately setting the radius of the tool path without damaging the cutting edge of the diamond is illustrated in Figure 8. First, a separate test part (A) whose width is smaller than the tool path diameter is mounted on the lens support fixture. A cylindrical surface (B) is cut by Tool path 1, and the X axis coordinate (X_1) is recorded. The X axis is jogged to the right and the opposite surface (C) is cut by Tool path 2, and its corresponding coordinate (X_2) is recorded. The width of the part (D1) is measured with calipers. The distance (D2) between coordinates (X_1) and (X_2) plus the width of the part (D1) is equal to the present diameter of the tool path. The amount of correction in tool path radius is calculated and the tool is removed from its holder. The X axis is programmed to increment over the calculated correction distance and the tool is reinstalled as close as possible to surface (C) by viewing through a microscope.

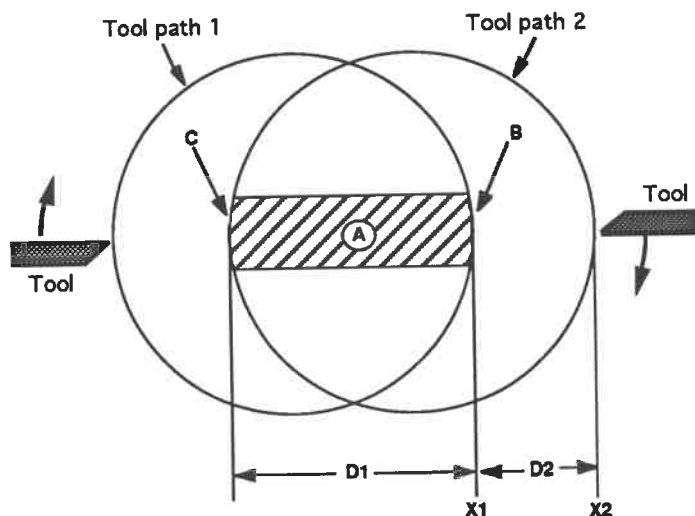


Figure 8. Technique for setting tool radius

Precision alignments

Fixture alignment and tool coordinate locations were important issues in the fabrication process. To insure accuracy, the Nanoform 600 was used in conjunction with an electronic pivot lever gauge as a coordinate measuring machine. The support fixture was aligned and shimmed for pitch, yaw, and roll with respect to the Z axis. This was accomplished by mounting the gauge on the X axis and scanning the mounting surfaces of the cantilevered support bar. At the same time, the bar height was adjusted to match the height of the optical axis of the lens to the spindle rotational axis. To maintain the same height of the optical axis of one lens surface with respect to its opposite surface, the same surface of each lens was always mounted on the support fixture between set-ups. To provide stability and minimize stress during interrupted cuts, the lens was clamped from the top with an elastic material between the clamp and the lens. This distributed the clamping force over the total area of the lens.

One important aspect of the fabrication process was the alignment of the optical axis in the Z direction for both toroidal surfaces on the same substrate. The Z coordinate of a previously diamond turned lens surface is lost any time the lens or diamond tool is relocated with respect to its support. Tool radius and lens position changes were required between setups of the different toroidal surfaces. A variation of a tool centering technique was used to relocate the Z tool coordinate with respect to the optical axis of the first surface. A small toroidal reference surface was cut after completing the first surface and before changing the setup. This reference was used to relocate the Z coordinate of the first surface before proceeding to cut the 2nd surface. A complete explanation of the "Chamfered Post Tool Centering Technique" is given in reference [4].

Conclusion

A method for producing large radius toroidal surfaces on a standard diamond turning machine has been successfully demonstrated. Convex, concave, and combination toroidal surfaces can be fabricated with small geometrical figure errors. For many applications, the resulting error is well within an acceptable range. Prototype lenses were diamond turned and used to verify critical optical designs before committing to polished molds for mass production. This technique is needed when the required radius of curvature is larger than the radial swing of the lathe. These surfaces would otherwise be impossible to produce without an extremely large radial capacity lathe. A special machine configuration with the spindle or the lens pivoting about an axis could conceivably continuously correct the cutting plane of the tool relative to the surface and eliminate all geometric errors. However, initial optical testing of these prototype lenses indicated that the quasi-toroidal surfaces produced exceptionally well focused beams.

References

- [1] Saito, T.T., Arnold, J.B., "Metrology of a Diamond Turned Toric Resonator Component", SPIE, Vol 93, 1976, pp 154-156.
- [2] Moriyama, S. et. al., "Development of a Precision Diamond Turning Machine for Fabrication of Asymmetric Aspherical Mirrors", Optical Engineering, Vol 27, No. 11, Nov., 1988 pp 1008-1012.
- [3] *Rexolite 1422 is a registered trademark for C-Lec Plastics, Inc.
- [4] Miller, A.C., and Cunningham, J. P., "Nanometer Tool Centering for Diamond Turning Applications", Proceedings from ASPE 1992 Annual Meeting, pp 216-219.

Acknowledgements:

This work was sponsored by LEXMARK International, Inc. Oak Ridge National Laboratory is managed by Lockheed Martin Energy Systems Inc. for the Department of Energy under contract DE-ACO5-84OR21400.

Molecular Dynamics Modeling of Thermal Conduction in Microcutting of Metals

Shoichi Shimada, Naoya Ikawa, Ryosuke Inoue, Junichi Uchikoshi

Department of Precision Engineering, Osaka University
2-1 Yamada-oka, Suita, Osaka 565 Japan

Background

Molecular dynamics (MD) computer simulation has been developed as an useful tool for analysis of microcutting process under ideal machine motion [1]-[5] . However, effect of thermal conductivity in metal has not been analyzed correctly in MD simulation of metal microcutting, because behavior of electron is not taken into account in conventional MD models. As thermal conduction in metals is mainly governed by electron mobility, metals in practice show considerably higher thermal conductivity than that estimated by the MD simulation. Therefore, for more realistic simulation of the microcutting process, it is advisable to take thermal conduction into account because chip formation is affected by thermal condition in shear zone.

Molecular dynamics simulation of thermal conduction in solid [6]

Figure 1 shows the 2-dimensional Morse potential type model used for the MD simulation of thermal conduction in a solid. To maintain the constant heat flux in the intermediate conduction layer, a certain amount of the thermal energy is added to all of the atoms in heating layer and subtracts the same amount in cooling layer, respectively, in the form of increments or decrements of both of potential and kinetic energy of the atoms at every computation time steps. The thermal conductivity of the solid in a steady state is calculated from the values of average temperature of atoms in the heating and cooling layers, T_1 and T_2 , heat flux, Q , geometrical dimension of the model, x and A , and the conduction period, t , as expressed in Eq.(1).

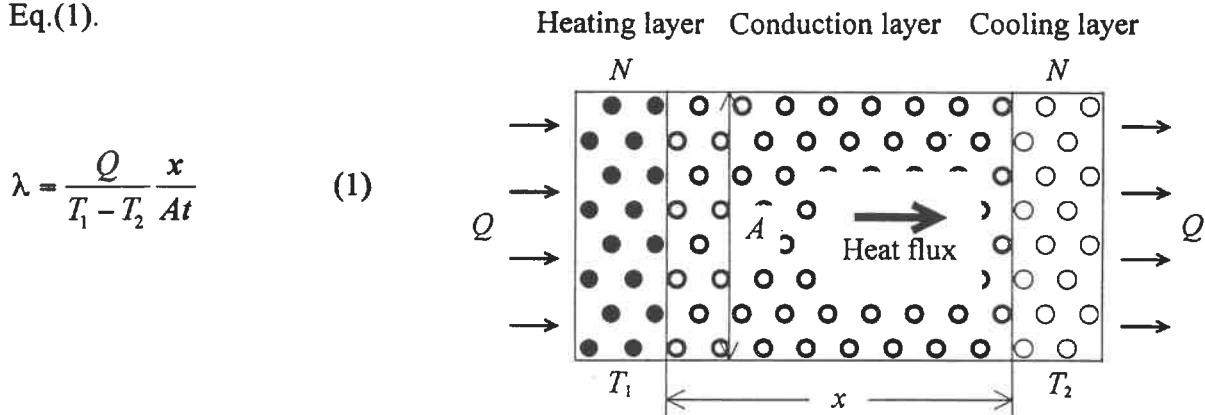


Fig.1 MD simulation model of thermal condition in solid

Thermal conduction modeling in metal

Theoretical thermal conductivity in solids, λ_t , follows the Eq.(2). The first and second terms are governed by the propagation velocity of lattice vibration and electron mobility, respectively. While heat in covalent solids is mainly conducted by means of lattice vibration, the proportion of electron conduction term λ_{te} exceeds 90% of the total thermal conductivity of metals λ_t . As the effect of this term is not taken into account in conventional MD simulations, metals in practice show considerably higher thermal conductivity and consequently smaller gradient in thermal field than those estimated by the MD simulations in terms of the lattice vibration. Therefore, for analyzing the thermal field in microcutting, the gradient in thermal field should be scaled so as to coincide with that obtained by continuum thermal conduction theory. The scaling is executed as is shown in Fig.2. Denoting the average temperature of atoms in the model by T_m , the scaled temperature T'_i of individual atoms which narrow the width of distribution, can be obtained by using Eq.(3). The constant β in Eq.(3) is so chosen that the heat conductivities coincides with the analytical prediction in a continuum model. The thermal conductivity of copper and aluminum obtained from the MD simulations shown in Fig.1 are summarized in Table 1 comparing with the theoretical and measured values. The result shows that the reasonable values of thermal conductivity can be obtained with the introduction of a suitable scaling for gradient in thermal field.

$$\lambda_t = \lambda_{tp} + \lambda_{te} \quad (2)$$

$$\lambda_{tp} = N_A k_B v_p l_p \cdots \text{Phonon conduction term}$$

$$\lambda_{te} = \frac{\pi^2 k_B^2}{3m_e} n \tau T \cdots \text{Electron conduction term}$$

N_A : Avogadro's number, k_B : Boltzman's constant,

v_p : Propagation velocity of phonon energy, l_p : Mean free path of phonon collision,

m_e : Mass of electron, n : Electron density, τ : Relaxation time

$$T'_i = T_i - \frac{(T_i - T_m)\beta}{k_B} \quad (3)$$

$$T_m = \frac{1}{N} \sum_{i=1}^N T_i$$

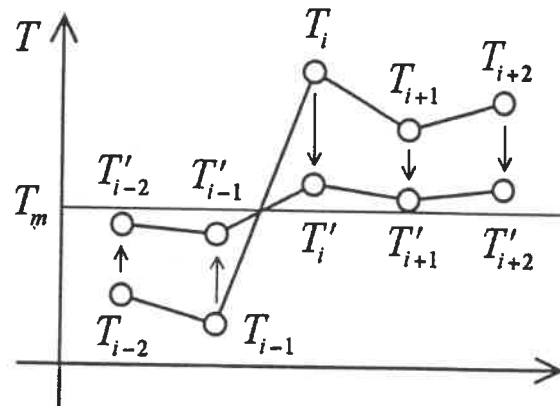


Fig. 2 Scaling of temperature distribution

Table 1 Comparison of thermal conductivity obtained by various methods

Material	MD simulation		Theoretical			Measured
	Non-scaled	scaled	λ_t	λ_{tp}	λ_{te}	
Cu	24	390	392	22	370	389
Al	15	230	231	16	215	233

(J / sKm)

Analysis of cutting temperature in copper microcutting

Figure 3 shows the difference in the equivalent temperature of atoms converted from kinetic energy between the thermal distribution obtained from scaled and non-scaled MD simulations. The results shows that chip formation process is significantly affected by thermal conductivity of workmaterial. The typical computation time is about 160 hours for 50 nm cutting distance on 12,000 atoms model at 20 m/s cutting speed on Sun SPARC station 10 Model 512. The average temperature of atoms in shear zone calculated from the simulation is 300 to 380K with introduction of the scaling proposed. This value is considerably lower than that (470 to 670 K) measured in the practical cutting experiments. However, uncut chip thickness in the cutting experiments are 2 to 100 μm , which are 2,000 to 100,000 times larger than that in the simulation. Therefore, the average cutting temperature in nanometric cutting may not rise so high as those in the cutting experiments. On the other hand, several atoms the equivalent temperature of which exceed 1,000 K are observed in the chip. The atoms at this temperature level can supply the necessary activation energy for chemical interaction with the oxygen in atmosphere. This result supports the fact that the surface of chips removed under nanometric uncut chip thickness is observed to be vigorously oxidized. Wear mechanism of cutting tool may also be understood from atomistic point of view based on the MD analysis of cutting temperature.

References

- [1] J.F. Belak, I.F. Stowers : A molecular dynamics model of the orthogonal cutting process, Proceedings of ASPE 1990 Annual Conference, Rochester, p.259, 1990.
- [2] I.F.Stowers et al. : Molecular Dynamics Simulation of the Chip Forming Process in Single Crystal Copper and Comparison with Experimental Date, Proceedings of 1991 ASPE Annual Meeting, Santa Fe, p.100, 1991
- [3] S. Shimada, N. Ikawa, G. Ohmori, H. Tanaka, J. Uchikoshi : Molecular dynamics analysis as compared with experimental results of micromachining, Annals of the CIRP, 41, 1, p.117, 1992.
- [4] S. Shimada, N. Ikawa, H. Tanaka, G. Ohmori, J. Uchikoshi, H.Yoshinaga : Feasibility study on ultimate accuracy in microcutting using molecular dynamics simulation, Annals of the CIRP, 42, 1, p.91, 1993.
- [5] S. Shimada, N. Ikawa, H. Tanaka, J. Uchikoshi : Structure of micromachined surface simulated by molecular dynamics analysis, Annals of the CIRP, 43, 1, p.51, 1994.

- [6] S.Kotake, S.Wakuri : Molecular dynamics study of heat conduction in solid materials, JSME International Journal, B-37, 1, p.103, 1994

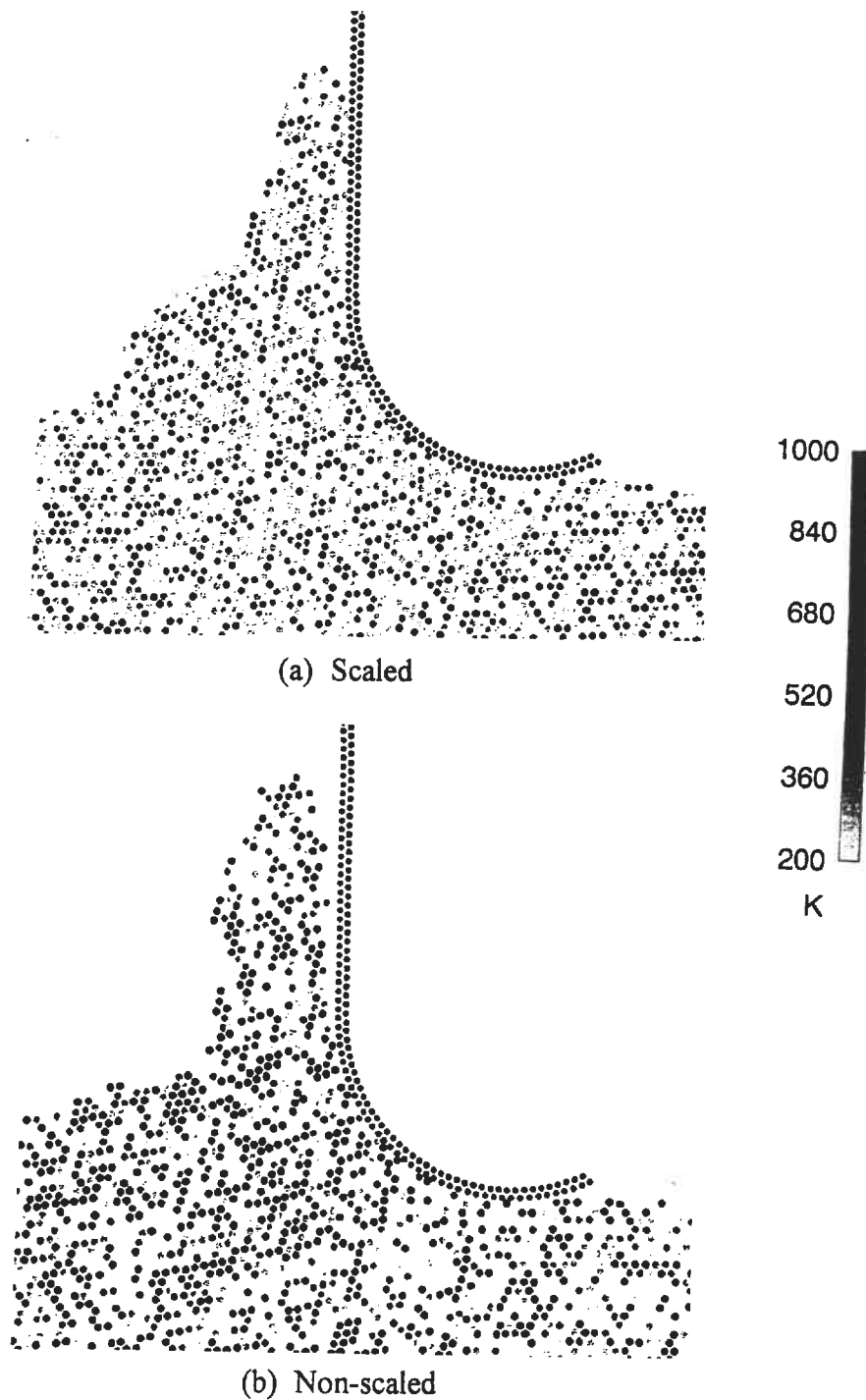


Fig.3 Effect of scaling of distribution of equivalent cutting temperature
Cutting speed : 20m/s, Cutting edge radius : 5nm, Uncut chip thickness : 1nm

Vibration Signature Analysis of AFM Images

G. A. JOSHI ^{§ †}, S. M. PANDIT [¶], and J. FU [§]

[§]National Institute of Standards and Technology, Gaithersburg, MD 20899

[¶]Michigan Technological University, Houghton, MI 49931

[†]Currently at Southern University, Baton Rouge, LA 70813

1 Introduction

Vibration signature analysis has been commonly used for the machine condition monitoring and the control of errors. However, it has been rarely employed for the analysis of the precision instruments such as an atomic force microscope (AFM). In this work, an AFM was used to collect vibration data from a sample positioning stage under different suspension and support conditions. Certain structural characteristics of the sample positioning stage show up as a result of the vibration signature analysis of the surface height images measured using an AFM. It is important to understand these vibration characteristics in order to reduce vibrational uncertainty, improve the damping and structural design, and to eliminate the imaging imperfections.

The choice of method applied for vibration analysis may affect the results. Two methods, the data dependent systems (DDS) analysis and the Welch's periodogram averaging method were investigated for application to this problem. Both techniques provide smooth spectrum plots from the data. Welch's periodogram provides a coarse resolution as limited by the number of samples and requires a choice of window to be decided subjectively by the user. The DDS analysis provides sharper spectral peaks at a much higher resolution and a much lower noise floor. A decomposition of the signal variance in terms of the frequencies is provided as well. The technique is based on an objective model adequacy criterion.

2 Experimental Detail

An experimental set up of the AFM, shown schematically in Fig. 1, was used to study the vibration characteristics of a sample positioning stage under a combination of a variety of suspension and damping supports. An experimental investigation was carried out using the following configurations (refer to Fig. 1):

Configuration 1: The sample was mounted on a solid metal block anvil placed on the metal base suspended on bungee cords. The AFM was then placed over the sample and was supported by the anvil.

Configuration 2: The tested invar positioning stage [13] was placed on the bungee cord suspended base on the air table. The AFM was mounted atop the invar stage over the sample.

Configuration 3: The invar stage and the AFM were mounted on a metal plate placed on top of the air table only.

Configuration 4: The invar stage and the AFM were mounted on metal bars placed on the air table only.

Configuration 5: A viton pad support was used underneath the tested invar stage placed on the bungee cord suspension kept on the air table. The AFM was placed on the invar stage. It is similar to the configuration 2 except for the three viton pads support.

The sample was held fixed, without XY scanning of the stage, through out the experiment. A 400 pixels $\times$ 400 pixels digital height image was collected in contact mode for each of these configurations. Each AFM image essentially measures the free vibration response of the sample scanning stage for different suspension and sample stage supports. Seven records of 400 data points each, from all of the five images are used to study the instrument response using DDS and Welch spectrum analysis.

3 Results

Five data records from the total 35 records taken for analysis, each containing 400 data points for configuration 1 through configuration 5 are shown in Fig. 2. A sampling interval Δ of 0.00122 seconds was used in each of these configurations. The mean signal level differs among the data records possibly due to variations in setting up the sample-tip contact mode. The amplitude and the nature of the vibration signal, however, are unique to each configuration. As seen in Fig. 2, vibration amplitude is the maximum for configuration 2, where the tested invar stage was set up on a bungee cord suspension. The vibration amplitude is the minimum for configuration 1 with sample mounted on a metal block anvil.

Welch periodogram spectrum estimation was performed for the same seven data records as used for the DDS analysis taken from every one of the five configurations. A rectangular window of 256 data points with no overlap and a total of 400 data points was utilized for calculation of the spectrum. A linear detrending was used before data windowing. The averages of seven spectra, one corresponding to each of the five configurations, are plotted in Fig. 3. Separate AR models were fitted to each of the seven data records from all five configurations using the DDS methodology. The averaged DDS spectra are presented in Fig. 4. Fig. 4 reveals these two resonant peaks at 120.31 Hz and 143.4426 Hz. The granularity of spectral calculations for DDS analysis was 0.8197 Hz. The averaged Welch spectrum (obtained using the same seven data records) shown as configuration 1 in Fig. 3 reveals only a weak resonance peak at 143.083 Hz. The 120.31 Hz peak is not present in the Welch spectrum plot. The granularity of Welch spectrum calculation was 3.2018 Hz. A finer frequency spacing cannot be used in Welch spectrum estimation without losing the smoothing effect (increasing the variance or unstability). The DDS spectrum calculation, in contrast, can be done at finer resolutions without such compromise, since it is based on the best fit ARMA model. The vibrational energy (area under the spectrum) for the configuration 1 spectrum is one of the lowest among the five configurations studied. However, a satisfactory vibrational performance on a metal block anvil is not a sufficient test; the sample scanning stage is required to carry out useful AFM imaging.

The adequate DDS models for the configuration 2 data records were found to be AR(6) using the model adequacy criteria for the nearly deterministic systems. One mode has a natural

frequency of 31.97 Hz and another mode with a natural frequency of 258.4 Hz. The spectrum plot in Fig. 4 reveals the peaks at these frequencies. It is important to attenuate low frequencies upto about 1 kHz affecting the AFM instrument such as in this configuration. These frequency modes are likely to have arisen from the lateral resonance of the tested invar stage. The averaged Welch spectrum plot shown in Fig. 3 configuration 2 shows matching peaks at 32.0184 Hz and 256.1476 Hz, respectively.

AR(14) model was judged adequate for configuration 3 data records using the deterministic system model adequacy criteria. The inspection of the DDS spectrum plot in Fig. 4 reveal six important frequency modes; the most prominent one has a natural frequency of 30 Hz. The averaged DDS spectrum for configuration 3 data records shown in Fig. 4 has peaks at 30.3279 Hz, 59.8361 Hz, 92.6230 Hz, 109.8361 Hz, 143.4426 Hz, 237.7049 hz, and 397.5410 Hz. The averaged Welch spectrum plot shown in Fig. 3 has matching peaks at 28.8166 Hz, 60.8350 Hz, 108.8627 Hz, 144.0830 Hz, 236.9365 Hz and 396.0287 Hz. However, the spectral peak at 92.6230 Hz is not present in the Welch spectrum of Fig. 3. Because of leakage and sidelobing it is difficult to estimate correct spectral peak strengths. In contrast, the DDS analysis reveals all the sharp peaks and the variance contribution estimates provide a quantitative measure of the peak strength. Spectral plots with possibly missing resonance peaks and disproportionate peak heights may mislead the experimenters, since the results are eventually used for design improvements and investigation of causes and cures to the instrument anamolies.

AR(22) model was found adequate for configuration 4 data records using the deterministic system model adequacy criteria. The averaged (from seven data record spectra) DDS spectrum for configuration 4 in Fig. 4 shows five peaks at 30.3279 Hz, 59.0164 Hz, 88.5246 Hz, 109.8361 Hz, 149.1803 Hz, and 237.7049 Hz. The averaged Welch spectral plot of Fig. 3 shows the matching peaks at 28.8166 Hz, 60.8350 Hz, 89.8516 Hz, 108.8627 Hz, 147.2848 Hz, and 236.9365 Hz, respectively. The peak heights and widths in Fig. 3, however, may not necessarily convey damping information or information about the relative strengths of the modes.

The averaged DDS spectral plot in Fig. 4 configuration 5 shows three peaks at 30.3279 Hz, 59.0614 Hz, and 289.3443 Hz. The averaged Welch spectrum shown in Fig. 3 shows matching peaks at 32.0184 Hz and 60.8350 Hz. However, the peak at 289.3443 hz revealed by DDS analysis is not present in the Welch spectrum plot. The 237 Hz resonance has been eliminated and 30 Hz peak is subdued in configuration 5, in which the viton pads support was used. This results from the good damping properties of the viton pads. Hence, configuration 5 gives the most desirable vibration free performance for the AFM operation.

4 Conclusion

Several data records from the AFM vibration images were considered for the study. The power spectral densities obtained by the DDS analysis and the Welch's periodogram method are compared. The DDS spectral plots revealed more peaks than the Welch's method in some cases. The DDS spectral peak location is estimated at a higher accuracy due to its ARMA model based super resolution estimation. The possible lateral resonant frequencies of the tested invar stage were found to be 30.75 Hz and 237.7 Hz. The damping provided by the viton pads was found to subdue these resonant frequencies. The use of bungee cord suspension without the viton

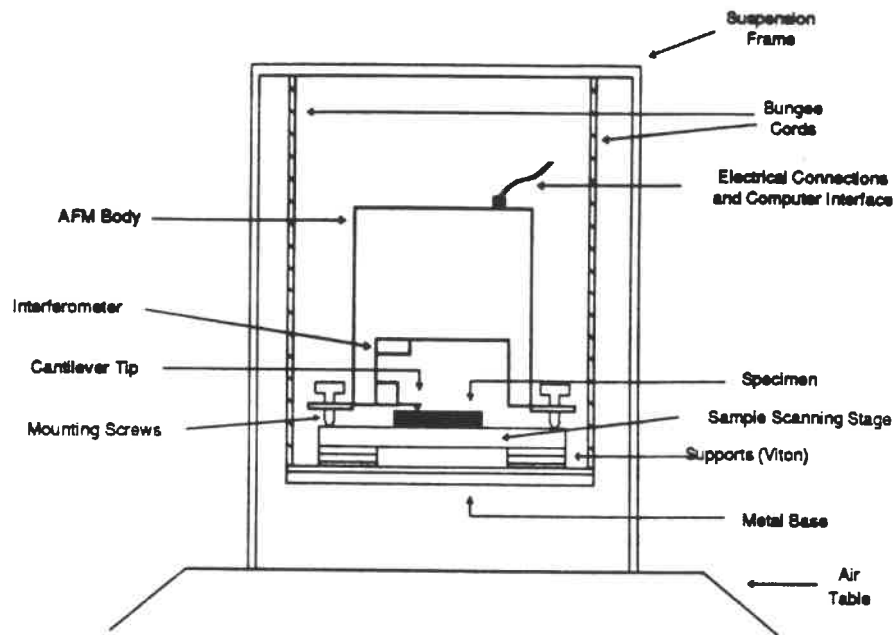


Figure 1: Schematic Sketch of an AFM

pad damping served as a vibration trap. The spectral analysis of the vibration records was found useful in the identification of the stage resonances and the understanding of the imaging imperfections.

5 Acknowledgements

This work was funded in part by the NSF grant #DMS-9223559. Partial support to GAJ from the Aerospace/Energy Research and Education Program (AREP) funded by NASA and DOE, grant #NAGW 3882, is thankfully acknowledged. Authors would like to acknowledge the guidance and technical contributions of T. V. Vorburger. V. S. Gagne drew the schematic sketch of the AFM.

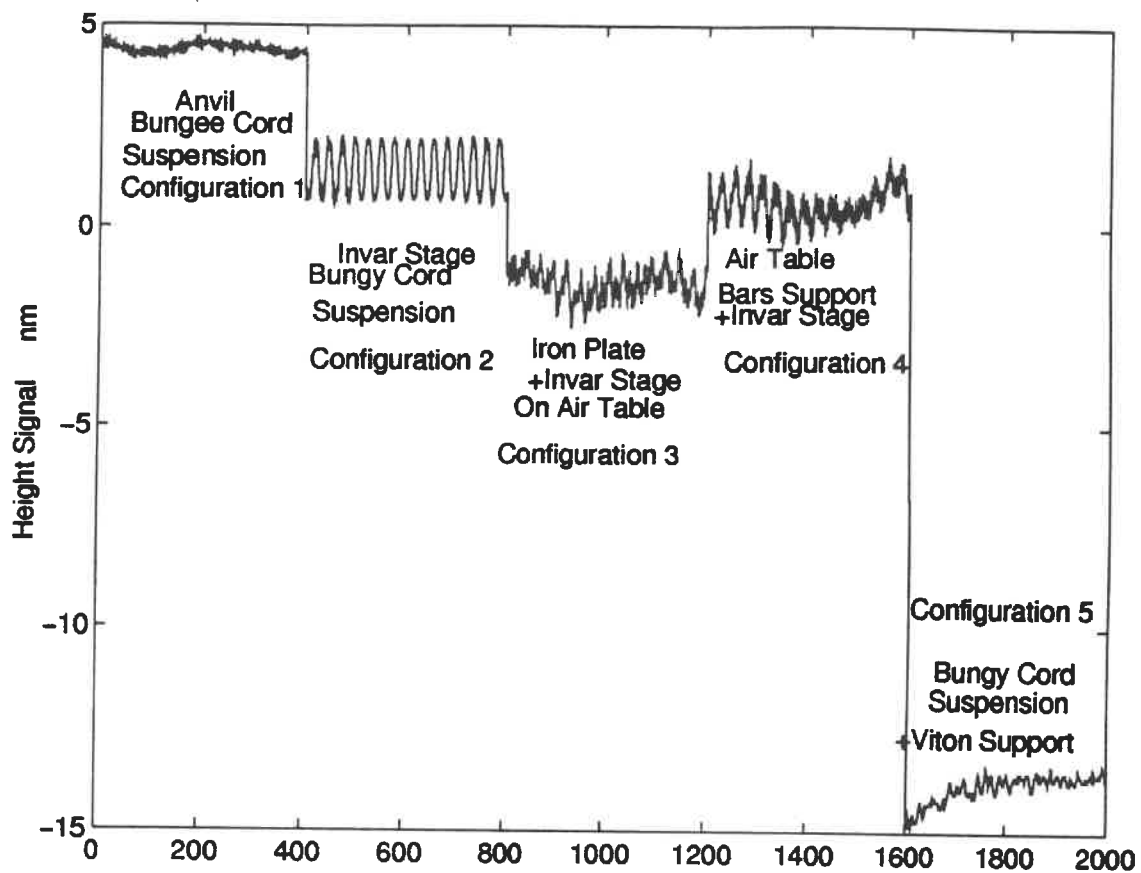


Figure 2: Free Vibration Records of AFM Tip. (The time scale for each data record is 0.49 seconds)

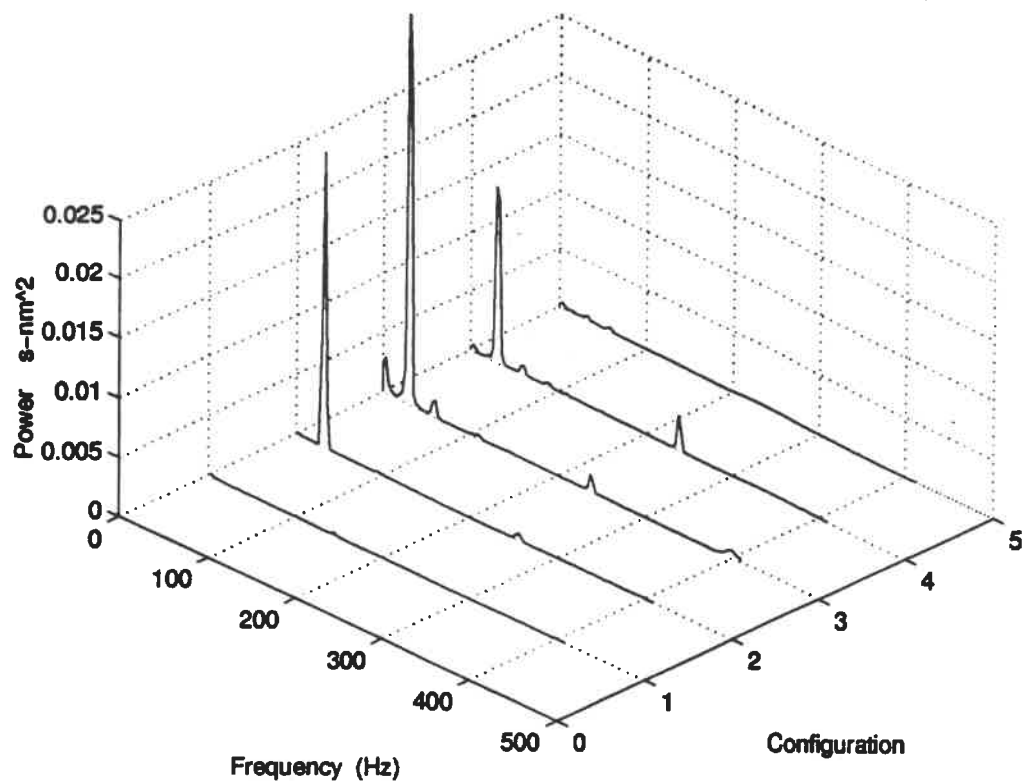


Figure 3: Power Spectral Density Estimates Using Welch's Method

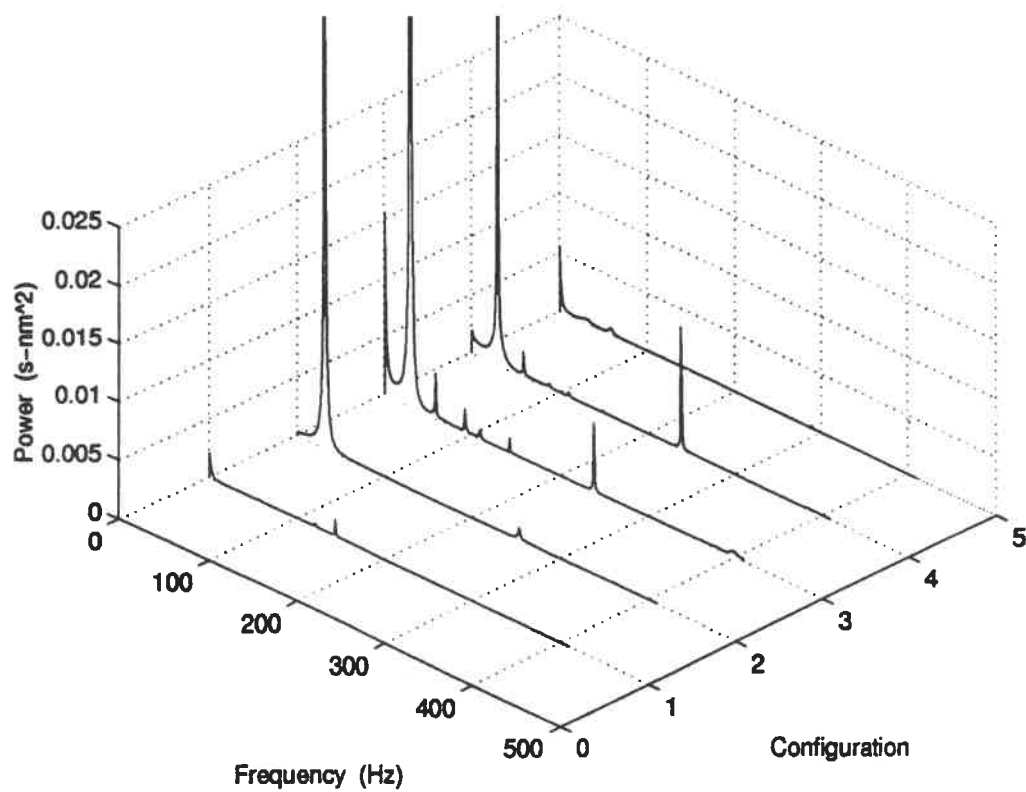


Figure 4: DDS Power Spectra of Vibration Records

A WAYLUBE PERFORMANCE COMPARISON USING A TELESCOPIC BALL BAR

**Tita Ouvreloeil, Philip McIntosh
CarboMedics™ Inc., Austin, Texas**

ABSTRACT

One parameter affecting machine tool accuracy is friction. Minimizing friction and friction force variability can improve machine performance. At CarboMedics™ Inc., a manufacturer of heart valves and related components, a periodic accuracy test of a CNC Milling machine using a telescopic ball bar showed a significant stick projection on Y axis. The machine manufacturer previously suggested that the problem might be alleviated by using a different type of waylube. A waylube comparison test was designed and performed to compare Mobil Vactra #2 and Sun 1180 waylubes, using a telescopic ball bar. Test data were analyzed using statistical methods. The t test showed no difference in roundness error, higher harmonic component, or stick slip between tests ($p=0.001$). However, the F test showed that Sun 1180 waylube gave better repeatability than Mobil Vactra #2 ($p=0.10$).

TEST METHOD

Stick slip can be reduced if conditions are such that friction increases as speed increases, instead of the more usual case of decreasing as speed increases. This requires that the coefficient of static friction divided by the coefficient of kinetic friction be less than 1, and the lower this value, the better.

The waylube used on this milling machine was Mobil Vactra #2 with a stick slip ratio of 0.71. The second waylube used for comparison was Sun 1180 with a stick slip ratio of 0.70.

The waylube test included a long-term warm up to insure, as much as possible, that the machine was thermally stable. For each test, data were collected for seven runs, in the XY plane, clockwise at 5 in/min using a telescopic ball bar (see Figure 1). The same coordinates were used for ball bar set up and the same operator performed all measurements. A thorough cleaning and pump-through of the lube system was done for each oil change. A reciprocal test was repeated after one month.

Roundness error was calculated using the Ball bar analysis software. Stick projection and higher harmonic component data were estimated by inspecting the plots (see Table 1)

Figure 1. Telescopic Ball Bar plot, clockwise direction, XY plane, 5 in/min

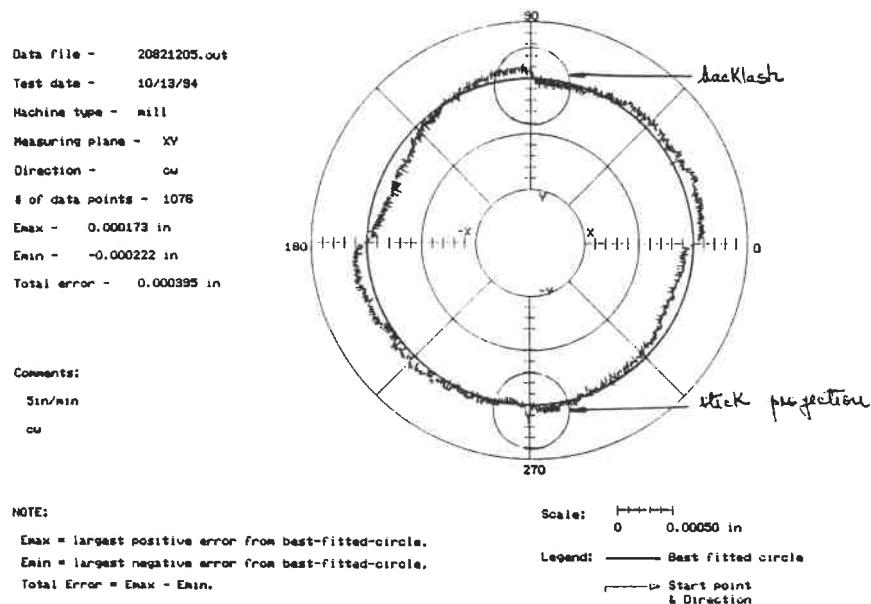


Table 1. Experimental data

Run number	Total error	roundness (µin)	Higher component	harmonic (µin)	Stick (µin)	projection
	Mobil V-2	Sun 1180	Mobil V-2	Sun 1180	Mobil V-2	Sun 1180
Lube Test						
1	576	527	120	100	200	300
2	506	501	120	100	200	220
3	504	510	110	110	200	250
4	509	507	100	110	200	250
5	598	529	110	100	210	250
6	493	501	100	100	200	250
7	513	520	110	100	350	250
Reciprocal Lube Test						
1	413	450	110	120	250	275
2	504	482	110	110	250	300
3	440	464	120	120	250	305
4	443	450	120	110	250	300
5	475	456	110	110	250	300
6	462	469	120	120	220	300
7	453	476	120	120	250	280

DATA ANALYSIS

1. COMPARISON OF TWO EXPERIMENTAL MEANS - T TEST

The t test is a method for comparing two experimental means. The null hypothesis, that the samples were identical and that the difference in the result $(\bar{x}_1 - \bar{x}_2)$, was the result of random fluctuations of the data, was tested. The test used the equation:

$$(1) \bar{x}_1 - \bar{x}_2 = \pm t s \sqrt{(N_1 + N_2) / N_1 N_2}$$

Where,

N_1 = number of degrees of freedom for the first oil ($N_1 = 7$)

N_2 = number of degrees of freedom for the second oil ($N_2 = 7$)

N = number of degrees of freedom for the system ($N = 14$)

$\bar{x}_1$ = mean of the first oil tested (see Table 2)

$\bar{x}_2$ = mean of the second oil tested (see Table 2)

s = pooled standard deviation for the test (see Table 3)

t = parameter depending of the number of degrees of freedom and the probability limit

Table 2. Means and standard deviations of the lube test

Calculation	Total error	roundness (μ in)	Higher component	harmonic (μ in)	Stick (μ in)	projection
	Mobil V-2	Sun 1180	Mobil-V2	Sun 1180	Mobil-V2	Sun 1180
Mean I	528.42	513.57	110.00	102.85	222.85	252.85
Mean II	455.71	463.85	115.71	115.71	245.71	294.28
St.Dev.I	40.97	11.77	8.16	4.87	56.18	23.60
St.Dev.II	28.81	12.57	5.34	5.34	11.33	11.70

2. COMPARISON OF PRECISION OF MEASUREMENTS - F TEST

The F test is a method for comparing the precision of two sets of measurements. The F test was also based on the null hypothesis that the precisions were identical. The test statistic F, defined as the ratio of the variances of the two measurements, was computed and compared with the maximum values of F expected (at a certain probability level) if no difference in precision existed between the two sets of measurements. The test used the equation:

$$(2) F = (s_1)^2 / (s_2)^2$$

where,

s_1 = standard deviation for the first test

s_2 = standard deviation for the second test

Table 3. Pooled standard deviations of the lube test

Pooled standard deviation	Test I	Test II
Total roundness error (μ in)	29.98	21.78
Higher harmonic component (μ in)	7.44	5.13
Stick projection (μ in)	44.23	27.52

RESULTS

The t test showed no difference in total roundness error, higher harmonic component, and stick slip between Mobil Vactra #2 and Sun 1180 waylubes ($p=0.001$).

The F test showed that the precision of the machine was improved when using Sun 1180 waylube ($p=0.10$).

CONCLUSION

The results suggest that a waylube change can be made, at nearly no cost, to improve machine repeatability. This improvement may be significant for operations requiring a particularly high precision. The results also indicate that the telescopic ball bar is a useful instrument for this type of comparison.

BIBLIOGRAPHY

1. Charles Lipson and Narendra J. Sheth, *Statistical Design and Analysis of Engineering Experiments*, McGraw-Hill Book Company, 1973.
2. Eugene Avallone and Theodore Baumeister III, *Marks' Standard Handbook for Mechanical Engineers*, McGraw-Hill Book Company, Ninth Edition, 1987.
3. Douglas A. Skoog, Donald M. West, and F. James Holler, *Fundamentals of Analytical Chemistry*, Saunders, 1988.
4. Cincinnati Milacron, *Lubricants*, 1985.

A 5 degree-of-freedom capacitive position sensor for straightness measurements

Jo W. Spronck, Marcus H.W. Bonse and Gert-Jan Nijssen

Laboratory for Micro Engineering

Department of Mechanical Engineering and Marine Technology

Delft University of Technology, Mekelweg 2, 2628 CD Delft, the Netherlands.

email: j.w.spronck@wbmt.tudelft.nl

Introduction

In many applications a precise linear movement of an object is desirable. Linear stages based on flexure hinges only can be used for relatively small displacements. For larger displacements a linear stage or bearing is employed. Realising a perfect straight motion of a stage or a linear bearing is an impossible task if we consider all possible undesired parasitic movements; displacements in two directions and three rotations: roll, pitch and yaw. These deviations from the linear motion can be due to manufacturing errors of the stage elements, thermal stresses or load changes. They can be reduced by calibration or feedback control of the critical element. For calibration or feedback control of a linear motion in x -direction, a sensor is needed that measures $d\varphi_x$, $d\varphi_y$, $d\varphi_z$, dy and dz ; 5 of the 6 degrees of freedom of an object. The coordinate system is defined in figure 1.

Up to now autocollimators, laser aligners and laser interferometers are used to measure one or more of these parameters (see also [1]). However, each of these systems can measure a maximum of two degrees of freedom simultaneously. It is necessary to use multiple, often different, systems to be able to measure all 5 degrees of freedom needed for straightness measurement. In this paper a capacitive position sensor is described that can measure all 5 degrees of freedom simultaneously relative to a straightness

reference using a single sensor structure [2].

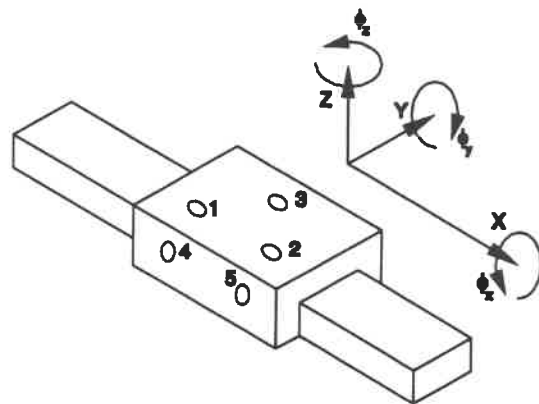


Figure 1: Definition of the coordinate system of a generalized straightness measurement system with the generalised position of the five position sensors used to measure the position of the object relative to the straightness reference.

Applications for a straightness sensor

Two different applications for this sensor can be discerned:

1. Calibration of conventional straight motion stages by the measurements of the five parasitic position errors, as shown in figure 2. All five degrees of freedom

can be measured in a single measurement run. Contrary to conventional measurement procedures, it is no longer necessary to change the measurement set-up to measure the different parameters of one axis.

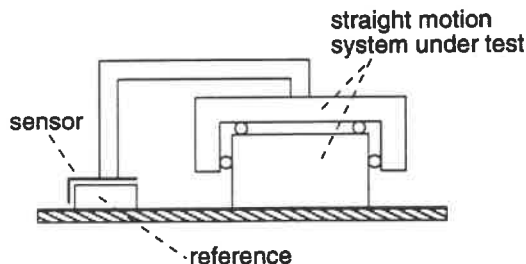


Figure 2: Example of a calibration set-up for a straight motion table by the use of the capacitive straightness sensor and a straightness reference.

2. Application in a feedback controlled straight motion system. As all five degrees of freedom can be measured simultaneously, it is possible to include the sensor in a feedback loop. Using suitable actuators, one or more of the deviations of the straight motion system can be corrected in real time.

The capacitive straightness sensor

The basic layout consists of a single sensor structure containing five capacitive detector-electrodes and one straightness reference electrode. All detector electrodes are in the same plane and can be manufactured as one part. High production tolerances are not required for the sensor electrodes only measure small changes in the capacitances withing a limited range. For this reason the sensor has not necessarily to be manufactured from a high-stability material. The reference electrode has to define a straight line and a plane; resulting into two types of straightness references:

1. A bar, consisting of one plane surface and one straight edge, as shown in figure 3.
2. Two flat electrodes, divided by a straight line or gap, as shown in figure 4.

A straightness measurement must measure the position of a sensor head with respect to a straightness reference in 5 degrees of freedom: three rotations and two translations. Assuming that the object to be measured is straight within certain tolerances, the measurement range of the 5 degree-of-freedom sensor has to be in the same order of magnitude.

Taking a prismatic bar, like a straight-edge, as the object to be measured it is possible to devise a sensor assembly to measure the necessary parameters. In figure 3 and 4 possible sensor structures is shown in which only capacitive position sensors are used. All capacitive sensors are in the same plane. To measure five degrees of freedom, five sensors are necessary, but more can be used.

Basically, three sensors are needed to find the z , φ_x and φ_y deviations. In figure 3 these are any three of sensors 1, 2, 3a and 3b. These sensors measure the distance of the electrode structure to the bar. Any arrangement of sensors not located on a straight line is sufficient, although of course the arrangement should be optimized.

For the y and φ_z measurement sensors 4 and 5 are used. These sensors are of a different type than sensors 1-3b, they operate on the edge of the bar, measuring lateral displacements in the $x - y$ -plane. Additionally, sensors 4 and 5 are sensitive to distance changes. As the capacitance values measured by the sensors are interdependent to a relatively large degree, meaning that for accurate straightness measurement calibration is necessary.

For calibration purposes a special straightness reference can be manufactured which allows better measurement of the relevant parameters by capacitive sensors. Such a system can be used for calibration of another straight motion

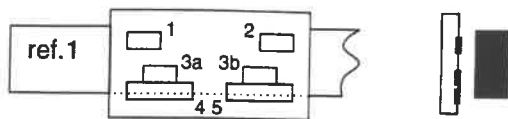


Figure 3: Cross section of the sensor and the bar-type straightness reference. This reference is electrically conducting and acts as the straightness reference. The reference straight line is defined as the edge of the bar. Using the measurement signals from the electrodes 1, 2 and 3, dz , $d\varphi_x$ and $d\varphi_y$ can be derived. Similarly the signals from the electrodes 4 and 5, above the edge, provide for dy and $d\varphi_z$. For experimental purposes an extra electrode has been added. Actually of electrodes 1, 2, 3a and 3b, only three are absolutely necessary.

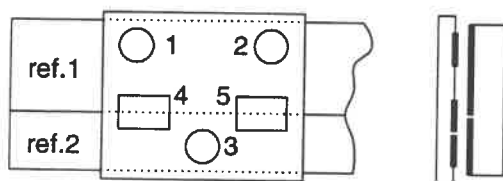


Figure 4: Top view and cross section of the 5 degrees of freedom sensor and the flat electrode straightness reference. Using the measurement signals from the electrodes 1, 2 and 3, dz , $d\varphi_x$ and $d\varphi_y$ can be derived. Similarly the signals from the electrodes 4 and 5, above the straight reference line, provide for dy and $d\varphi_z$.

system, for instance the guideways on a lathe or turning machine. An example of such a system is shown in figure 4.

This configuration, where not a bar but a custom-made reference electrode is used, is better suited for measuring y and φ_z deviations due to the better defined capacitance change due to these deviations. Using a suitable measurement strategy, the two sensors are not sensitive to distance changes, but only to lateral displacements.

The straightness reference is a form standard. The accuracy of the straightness measurement system is limited by the accuracy of this form standard, assuming the measurement system is ideal. It is clear that the straightness reference has to be manufactured from a high-stability material.

Results

Using an experimental set-up according to the principles as shown in figure 3 and 4, submicrometre repeatability has been achieved with a 1 mm gap between the sensor and the reference electrode.

The measurement electrode used was manufactured from printed circuit board. The size of the individual electrodes was chosen in such a way that the nominal capacitance in all cases was 1 pF. The total size of the electrode structure for the sensor type shown in figure 3 was $80 \times 25 \text{ mm}^2$, for the type shown in figure 4 $100 \times 80 \text{ mm}^2$.

The straightness reference was an optical flat with a metal layer deposited on one side.

The electronic measurement system was based on the modified Martin-oscillator and had the capability of measuring up to eight different capacitances [3]. The linearity of the electronics is better than 10^{-4} between 0–4 pF. The electronic measurement system is equipped with a microcontroller and serial communications port.

The data were sent to a personal computer, where (using the program HP-VEE) the position information was extracted from the capacitance data. It is also used to translate the position information measured by the sensor to the straightness information about the object being measured (see figure 2).

The results of resolution and stability measurements on the straightness sensor are shown in table .

coord.	resolution
y	0.2 μm
z	0.1
φ_x	3 μrad
φ_y	3
φ_z	5

Table 1: *Results of resolution and stability measurements on the capacitive straightness measurement system. The distance between sensor and reference was 1 mm. Differences in resolution occur due to the different kinds of sensors and algorithms used to find the position change from a capacitance change.*

The linearity of the straightness measurement was better than 1% of full scale for all coordinates, the range for y and z was 0.25 mm, for φ_x , φ_y and φ_z : 2 mrad.

Conclusions

The capacitive straightness measurement system described can measure 5 degrees of freedom simultaneously. It is easy and flexible in design and production. Especially the fact that the principle is scalable creates a wide field of applications. The electronic measurement system as used in the experimental set-up is a low-cost implementation. The sensor electrode structure itself is not a critical part, only one calibration with the reference (which is a critical part) is needed for absolute 5 degree-of-freedom measurements. Due to the relatively small measurement range necessary for

a straightness measurement system the resolution and accuracy are high.

Typical applications for this sensor are:

- A calibration facility for linear stages.
- Use of the sensor in a feedback controlled straight motion system.

References

- [1] Shimokohbe A and Aoyama H. An active air bearing: a controlled-type bearing with ultra-precision, infinite static stiffness, high damping capability and new functions. *Nanotechnology*, 2:64-71, 1991.
- [2] Bonse M H W. *Capacitive position transducers; theoretical aspects and practical applications*. PhD thesis, Delft University of Technology, 1995.
- [3] Toth F N, Meijer G C M and Kerkvliet H M M. Ultra-linear, low-cost measurement system for multi-electrode pF-range capacitors. In *Proceedings of the IMTC/95*, pages 512-515, Boston, MA, April 24-26, 1995.

Autonomous Calibration of a Hexapod Machine Tool

Hanqi Zhuang
Department of Electrical Engineering
Florida Atlantic University
Boca Raton, FL 33431

Abstract

In this paper, an autonomous approach to the calibration of a new-type of CNC machine tool is proposed. It is shown that by installing 6 redundant sensors on the hexapod machine tool, the system is able to perform self-calibration. The self-calibration algorithm for the hexapod machine tool has the following additional advantages: (1) It does not need to solve numerically the forward kinematic solution, and thus improves the robustness and reduces the computational complexity; (2) It converges very rapidly; (3) Its accuracy performance under noisy measurements is satisfactory; (4) Only about 12 measurements are needed for a robust calibration of the system; and (5) This sensor-placement strategy has the advantage of balancing the loading effect of the system. The proposed approach provides a tool for rapid and autonomous calibration of the parallel machine tool.

Summary

A new class of CNC machine tools based on an inverted Stewart platform is emerging. It promises greater accuracy if properly calibrated. The objective of this research is to develop an autonomous, on-line calibration technique for such parallel-link machine tools.

Of all machine tools, CNC machine tools are the most important ones. They are very flexible and versatile in machining geometrically complicated parts. Further, the ever increasing computational power of CNC controllers allows the user to enhance the machine

accuracy by periodic execution of special calibration software routines, using appropriate sensing devices.

The geometry of CNC machine tools is typically open-chain. A new machine tool, developed at Ingersoll Milling Machine Company, is based on an inverted Stewart platform. Unlike conventional machine tools, the Ingersoll machine has a fixed table that carries work pieces. An octahedral hexapod with the tool is hung above the table. The hexapod, moving with 6 degrees of freedom (DOF), has the potential of machining to micron accuracy if properly calibrated. The manufacturer claimed a repeatability of better than 1 micrometer. The repeatability was maintained when the machine tool operated under various loading conditions. This feature provides a good potential for machine calibration. It is a process by which the actual kinematic parameters are estimated and then used to modify the kinematic model residing in the manipulator controller. Kinematic calibration can significantly improve the manipulator accuracy.

The calibration of this type of machine tools, however, is a challenging task. While there are numerous reports dealing with the calibration issue of open-link machine tools, the investigation of parallel-link machine tools is in its infancy. The workspace of this machine tool is very limited and the structure of the hexapod is very large, therefore it is difficult to measure the machine tool's geometry by a coordinate measuring machine. Although theoretically a laser tracking measuring system can be used to measure its geometry, either it does not have sufficient accuracy or it is too expensive. Moreover, the machine tool may not be calibrated on-line as the laser beams may be blocked by work pieces.

Machine tool calibration is normally done in a well-controlled laboratory environment. Pose data of the manipulator is measured by a dedicated measuring device. This, together with internal sensing data, is used to identify the kinematic model or model parameters of the system. However, accurate calibration data through external sensing is expensive and difficult to obtain. For a system that functions outside of a controlled laboratory environment, it would be desirable not to use special-purpose calibration equipment to update the model used for system control. It is thus essential that the system is able to perform self-calibration.

Autonomous calibration has the potential of, (a) removing the dependence on any

external pose sensing information, (b) producing high accuracy measurement data over the entire workspace of the system with an extremely fast measurement rate, (c) being automated and completely noninvasive, (d) facilitating on-line accuracy compensation, and (e) being cost effective.

A self-calibration task may be performed by imposing physical constraints to the system or by adding redundant sensors. Calibrating a laser tracking system by restricting the target motion on a planar surface is an example of the former, and using redundant sensing information to calibrate a machine tool is an example of the latter. This paper concentrates on the second approach.

It is illustrated through numerical study that by placing 6 additional sensors on 3 alternative U-joints of the Stewart platform, the system is capable of performing self-calibration. This sensor-placement strategy has the advantage of balancing the loading effect of the hexapod machine tool.

The key step in mathematically formulating a traditional kinematic calibration problem is to derive a measurement residual, which is the discrepancy between the measured and the computed manipulator poses (or their equivalents). For instance, in the case of calibrating a serial manipulator, the measurement residual is normally the measured pose and the pose computed using the nominal robot kinematic parameters. For calibration of Stewart platform, rather than using the pose error as the measurement residual, one can use the discrepancy between the measured and the computed leg lengths.

In the case of self-calibration, construction of a measurement residual is not a trivial task since no information about poses of the manipulator is directly available. We overcome this difficulty by exploring all possible information provided by redundant sensing.

A parallel-link machine tool has its forward and inverse kinematic solutions without redundant sensing, although in some cases these solutions can only be obtained by numerical means. By installing redundant sensors, one may be able to find the manipulator's forward and inverse kinematic solutions through a different path. Ideally, the solutions provided with and without redundant sensing should be consistent. However, because of the errors in the nominal kinematic parameters of the manipulator, these solutions from two paths may not be the same. The discrepancies between these solutions can then be used to form measurement

residuals. If the joint variables are sensed at a sufficient number of measurement configurations, the actual kinematic parameters may be estimated by minimizing the measurement residuals and be used to modify the kinematic model resident in the machine controller.

The demand for higher quality products and better efficiency of manufacturing essentially requires higher accuracy of machine parts in dimensions and surface texture. Toward this goal, there is a need to enhance the accuracy of the machines by which these parts are fabricated — CNC machine tools are among the most important ones.

The geometry of CNC machine tools is typically open-link. A new class of CNC machine tools based on a Stewart platform promises better accuracy if properly calibrated. In a conventional calibration method, machine poses need to be measured with an external device. However, this task is very difficult to perform for the new class of machine tools. The proposed self-calibration technique will radically alter the way calibration is done. It will make external sensing unnecessary.

A major feature of this approach is the addition of a few redundant internal sensors to the machine tool. These can be integrated into the initial machine design. For high performance CNC machine tools, the proposed technique is cost effective. It is possible to keep the cost of additional internal sensors to within a small fraction of the total price tag of the system. In return, the system will be able to automatically calibrate itself.

With the proposed approach, the machine tool can also do self-calibration while it performs normal tasks. This will minimize the system service time and improve its productivity. Thus, the economic benefit of this approach is significant.

As a final remark, the on-line calibration and compensation strategy can be applied to other types of CNC machine tools as long as pose errors of the machine tools can be measured, directly or indirectly. This will also have a very positive impact on the U.S. machine tools industry.

Using Capacitance Probes to Measure the Limit of Machine Contouring Performance

Don Martin

Lion Precision 563 Shoreview Park Road St. Paul MN 55126

Most machine tools used for discrete part manufacturing have the ability to create contoured surfaces. Generating a contoured surface usually requires the coordination of two or more axes of the machine. The increased use of CAD/CAM and integration of engineering and manufacturing systems has put increased pressure on contouring accuracy. This paper examines the nature of contouring errors, and presents a simple approach using capacitance displacement probes to determine the contouring speed limit of a machine tool.

Contouring Error and How It Is Measured

A machine is programmed to travel in a predetermined path or contour. Any deviation from the programmed path is referred to as contouring error. For simplicity of implementation and analysis, most contouring tests use a circular path. A machine is programmed to travel in a circle, the resultant path is measured and the difference between the programmed path and the actual path is compared. Not only is the error of interest, but by studying the characteristics of the error, many times the cause can be found and the problem eliminated. ANSI/ASME B5.54 "Methods for Performance Evaluation of CNC Machining Centers" recommends the use of the telescoping ball bar to measure contouring performance of a machine tool. The method presented in this paper is an alternative to this approach and it allows an arbitrary path and much smaller distances.

What can be done to push a machine to its (speed) limit?

All machines have an upper limit on contouring speed. It is related to how fast the machine can accelerate (servo response) and how fast the processor can calculate target positions. It makes sense that the ideal contouring test would require a situation with maximum accelerations. One approach to generate demanding accelerations and high calculation rates is to program the machine to contour a very small circle. Even at moderate speeds, tracing small circles is a very difficult test for a CNC machine tool.

Consider what is going on with two axes that are performing a circular contour motion: Each axis is being driven with a sinusoidal function. The position, velocity and acceleration formulas for an 'x' and 'y' axis are:

$$\begin{array}{lll} x = K \sin(\omega t) & v_x = \omega K \cos(\omega t) & a_x = -\omega^2 K \cos(\omega t) \\ y = K \cos(\omega t) & v_y = -\omega K \sin(\omega t) & a_y = -\omega^2 K \sin(\omega t) \end{array}$$

At a feed rate of 250mm (10 inches) per minute and a radius of 250mm (10 inches), the maximum acceleration works out to be 0.000 069m/s² or 0.000007 g_a

For each order of magnitude of reduction of radius, the acceleration increases by one order of magnitude (if the speed remains constant).

By changing the path diameter from 250mm to 0.25 mm and using a constant feed rate acceleration increases by a factor of 1,000. From a technicians standpoint, I can assure you it is much more comforting to be observing a machine going in 0.25mm circles at 250mm/minute than one going in 250mm circles at 250,000 mm/minute!

From the above analysis, it is apparent that by reducing the path diameter, it will be possible to determine the limitation of a high speed machine tool by measuring over a small displacement range.

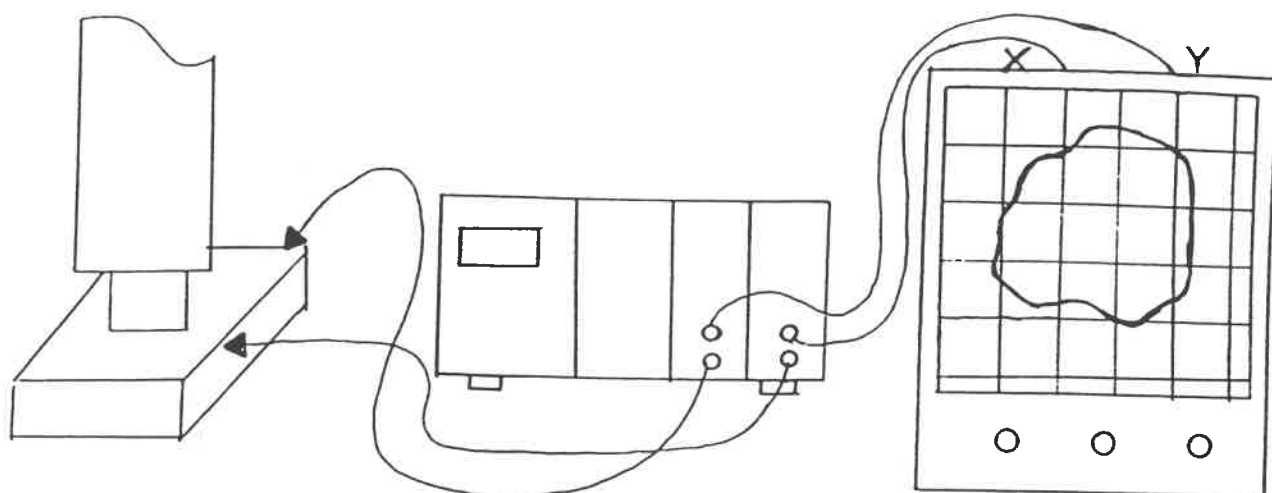
Test setup

Whenever possible, measurements on machine tools should simulate measurement at "the point of machining" which represents motion between the "tool" and the "workpiece" during normal operation. On a machining center this involves mounting probes on the machine table and mounting a target on the spindle or spindle housing.

The setup for this test is shown in Figure 1. A square target is mounted to the spindle so that it does not rotate and is aligned in the X and Y directions. Capacitance probes are fixtured to measure along the X and Y edges of the square. The input of the probe drivers is output directly to an oscilloscope where the machine tool motion is directly displayed (and printed). Figure 2 shows more detail of the probe and target mounting. The "square" was a piece of precision ground steel, the probes, probe drivers, and mounting fixture are part of a commercial spindle analyzer from Lion Precision, and the oscilloscope was a commercially available unit from Fluke.

The machine tested was in good operating condition and was used in regular production. It is a Matsura MC760V manufactured in 1982. It is specified to have 118 inches per minute linear interpolation and 40 inches per minute circular interpolation capability. The machine was programmed using the circular interpolation command, using various path diameters and speeds.

CAP GAGE CONTOURING TEST FOR MACHINE TOOLS



Machine Spindle	2 Channel Cap Gage	Oscilloscope
Target	Measuring Range	w/ X-Y Display
Two Cap Gages (X,Y)	0.060 - 0.080 Inches	Storage/Print Capability

Machine Tested:

Matsura MC760V 1982

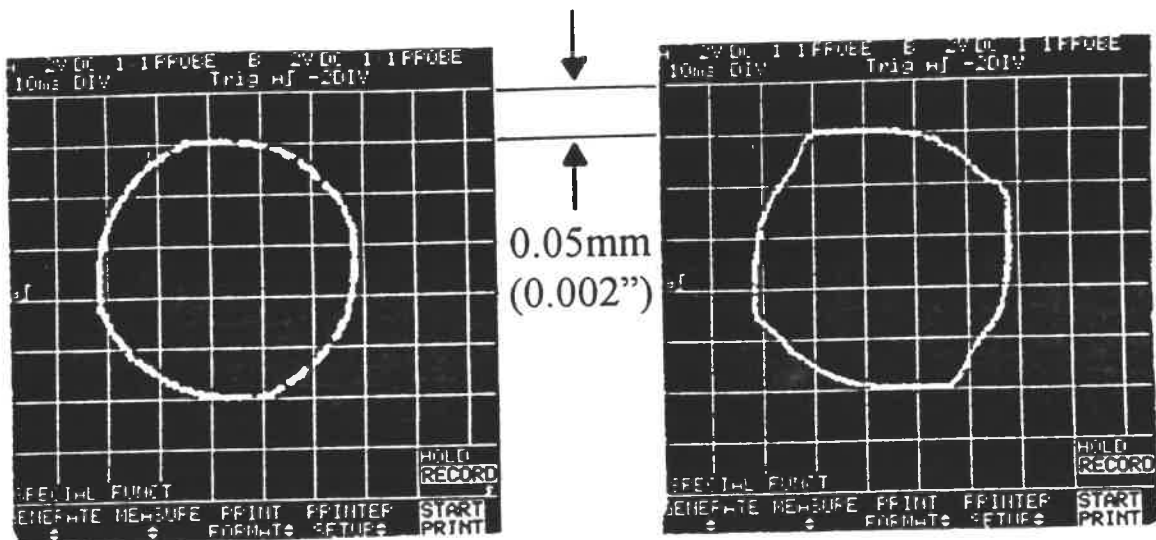
118 Inches/Minute Linear

40 Inches/Minute Circular Interpolation

Figure 1
Instrumentation Test Setup

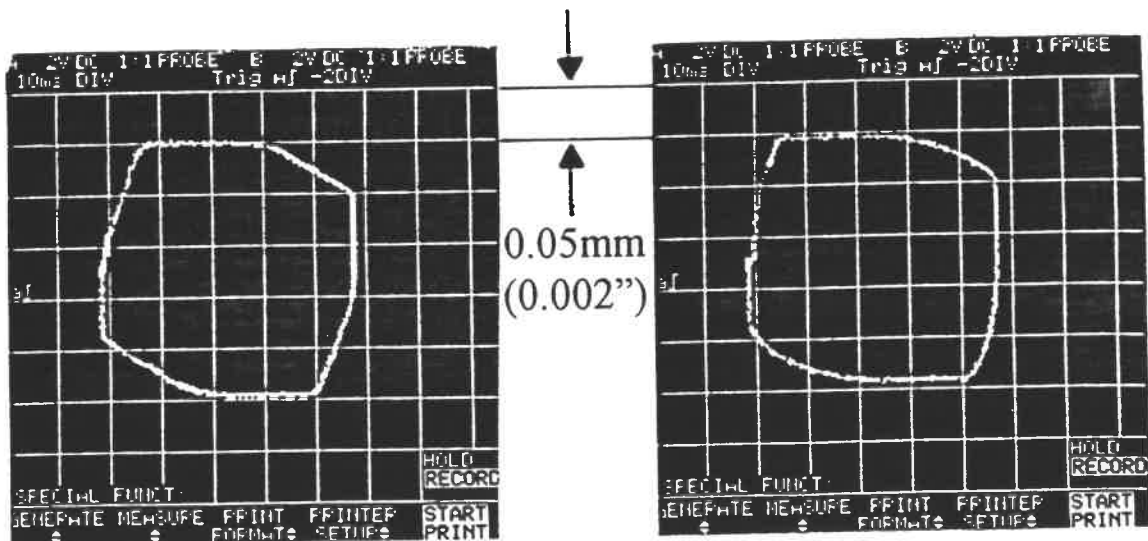
CAP GAGE CONTOURING TEST FOR MACHINE TOOLS

PATH DIAMETER 0.010 INCHES



2.5mm/min (0.1 IPM)

25mm/min (1 IPM)



50mm/min (2 IPM)

125mm/min (5 IPM)

Conclusion

This test is very capable of finding the limit of contouring speed for a machine tool.

Advantages

Test set-up is simple

CNC programming is trivial

Capable of finding limits of performance

Very safe and low-key compared to other "speed limit" tests

Accommodates small work zones

Works at speeds up to 10,000 circular contours/second (1,000,000 inches per minute for 0.5 inch diameter or 10,000 ipm for 0.005 inch diameter path)

Provides calibrated, traceable output

Disadvantages

Short range: 0.5 inches is maximum practical range

Requires target mounted in or on spindle

Further Research and Application

This capacitance sensing approach allows for arbitrary paths - the test setup does not dictate a circular path. This fundamental difference from current contouring tests such as the telescoping ball bar and the circle trace test provides more flexible testing and shorter set up times. In addition a third probe could be mounted in the 'Z' direction and position in all three directions detected simultaneously.

This same concept allows for use of plane mirror interferometry for larger ranges. Again X, Y, and Z position could be detected simultaneously. This would provide interferometric accuracy, speed and range with arbitrary path capability.

Summary

A method has been presented to determine the limit of contouring speed on CNC controlled machine tools. All the instruments used to make the measurement are readily available and straightforward to apply. Results of measuring a typical machine tool are presented. Advantages, disadvantages, and further application of the measurement technique are discussed.

Acknowledgements

Robert (Buz) Callaghan for discussions on the subject in 1993, encouragement from James B. Bryan to perform the tests and present the results, and Greg Vogel of Professional Instruments Company for providing machine time, fixturing, and instrumentation.

A Study on Straight Movement Error Detection in the Method of Laser Datum

Wan De-An Gao Yingjie Li Yan Ma Shumei
Mechanical Engineering Department
Tong-Ji University, Shanghai 200092, China

1. Introduction

In order to meet the needs of the development of micro-electronics, micro-mechanics and optical technique, the nanometre machining and measuring technology attracts more and more attention. Because the typical measuring system, such as slide guide, level pressure guide or air guide, adopts geometrical datum, it is very difficult to obtain a higher accuracy than $0.1 \mu\text{m}/100\text{mm}$. The machining error, the clearance of the kinematic pair, the force and the thermal deformation of the system themselves all will effect the stability of geometrical datum. At the time when people work hard to improve machining accuracy of all sorts of guide, they also begin to develop many error separation technique. As the circular movement of spindle has an excellent cyclicity, the error separation method has successfully used in the roundness measuring, and nanometre order accuracy has been attained. Someone has adopted inverse measuring method, two-point method or three-point method to deal with the mixed error of the straight movement and the shape, it is hard for them to obtain a nanometer accuracy. As the straight movement doesn't possess the strict cyclicity, the straight movement error and the shape error can't be effectively separated from each other. Japan scientist Hideki Matsumoto et al [1] suggest a real measuring and controlling method to compensate for the movement error, and have obtained a position accuracy of $0.05 \mu\text{m}$ referring to the reference datum within 600 mm displacement. Because of the machining error of the reference datum itself, this method only attained $2 \mu\text{m}$ absolute accuracy finally.

Here we suggest a new measurement method using laser as the physical datum of the straight movement. This method can avoid the influence of various errors of geometrical datum. If only the effect of thermal deformation of light source and change of environment on the laser beam can be controlled, we can keep the direction of laser beam stable. Shape error measuring system using such light-beam as straight movement datum can attain nanometer measuring accuracy.

2. Controlling of the stability of the laser beam direction

According to the zero-position principle, the influence of movement error can be avoided by feedback control when the Quadrantal Photodiode is mounted on the straight movement object and used as 2-D spot position detector, this can refer to reference [3]. But as the datum, the stability of the laser beam itself is very important.

2.1 Protection of beam drift caused by thermal deformation

The direction of laser beam will drift as the heat of the laser source can result in thermal deformation. The drift will seriously hold up the application of the laser datum. Here we suggest a new method to control the stability of the laser beam's direction. As shown in Fig.1, He-Ne laser is mounted on a elastomer which is supported by two piezocrystal (PZT). The laser beam is separated into two branches by a splitter: one, through the splitter, illuminates the QPD1 and is adopted as the straight datum; the other illuminates the reference detector QPD2. Both QPD1 and QPD2 will detect the partial difference caused by the laser beam drifting. After QPD2 obtains the signal of

photocurrent difference caused by position difference, the signal can be processed and transported to PZT which is mounted on the laser source system. So the compensation is realized, zero-position is kept and the stability of laser beam is guaranteed.

Fig. 2 is the real-measuring figure of laser beam's stability without the feedback control; that the horizontal resolution is 2 nm, the vertical's is 3.8 nm. Within the optical distance $L=500\text{mm}$, 20s interval, the drift of laser beam is $1.3\text{ }\mu\text{m}$ ($2.6\times 10^{-6}\text{rad}$) horizontally and $2.5\text{ }\mu\text{m}$ ($5\times 10^{-6}\text{rad}$) vertically. Fig. 3 is the result of adopting the feedback control. The drift of laser beam is $0.20\text{ }\mu\text{m}$ ($4\times 10^{-7}\text{rad}$) horizontally and $0.36\text{ }\mu\text{m}$ ($7\times 10^{-7}\text{rad}$) vertically within two hours. So the new method, which adopts the feedback control to improve the laser beam's stability, is feasible.

2.2 Protection of air fluctuation in optical path

The air fluctuation is another fact that will effect the stability of laser beam. There is a transmission distance between laser source and the photoelectric detector QPD. The variation of air density, pressure and dust, humidity and temperature may greatly influence the stability of laser beam, and the air flow, vibration, noise of the surrounding all will result in the fluctuation of laser beam too. Because the influence to the stability of laser beam is very complex, we dealt with our analysis and calculation, simulated experiment and real measurement only about the variation of temperature, pressure and humidity on a special condition. We conclude that even in the constant temperature room, the fluctuation of laser beam is $1\times 10^{-7}\text{rad}$ or so caused by changes of air density etc. A vacuum cover was adopted in our experiment. It is very difficult to keep vacuum working state for a long time between the kinematic pairs. Here we use the method with an axle embedded in a hole. The vacuum is 0.01 barometric pressure. This structure can keep a pressure at 0.01 barometric in the vacuum tube within 2 or 3 hours, and the stability of laser beam is about $5\times 10^{-8}\text{rad}$.

3. Straight movement system

The straight movement system of laser datum consists of a laser source and its fine adjustable platform, a straight movement platform, a displacement sensor and its fine driving system, feedback control system and antivibration platform. This apparatus is about 1400 mm long, 650 mm wide and 400 mm high on the whole, and can be fixed on the 5-foot-long large antivibration platform.

3.1 Adjustable system of laser beam direction

In order to detect shape error with the laser datum, first, the laser beam and the moving direction of the straight movement platform must be adjusted to be parallel with each other by the adjustable system. This can keep the laser spot spotting within the measuring range of the photoelectric sensor and realize the stretching out and the drawing back compensation. The maximum of the stretching out and the drawing back of such cascade piezocrystal we used is $\pm 16.5\text{ }\mu\text{m}/\pm 150\text{V}$. Second, the laser beam's centre and the QPD's can be adjusted to be coincide with each other using this adjustable system. This is the guarantee to realize zero-position detection and control. The adjustable system we developed is simple but useful. It is a microadjustable elastic structure, made of casting pig, and finished by elastic sparking. Four piezocrystal are fixed on both the back and forth carriage of the elastic structure to compensate for the drift of the laser source caused by the thermal deformation. The real measuring result told that the resolution of this adjustable system could attain 0.012nm both horizontally and vertically.

3.2 Shape error measuring system

Photoelectric detector QPD and the stretching out and drawing back elastic structure

composed of piezocrystal are mounted on the straight movement structure of the shape error measuring system. This design can keep the measuring sensor move along with the laser beam with high accuracy. In order to scan the whole shape error of workpiece line by line, the workpiece to be measured is fixed on the table which can move crosswise. The system is made of aluminium alloy, finished precisely by electrosparking. Position sensor are fixed on it and can use piezocrystal on the system as micro-driving compensation system.

4. Test result with laser datum: shape error measuring of optical flat

The optical flat is usually used as measuring datum, it possesses a very high shape accuracy. So it is very difficult to measure its shape error. A high sensitivity circular current sensor we made ourselves is mounted on the straight movement structure, and for the convenience of measuring, a very thin layer white silver alloy is electroplated to a flat of 50 mm in diameter, 30 mm in thickness evenly.

Fig. 5 is the test result of the flat without using laser datum, and curve 'a' is the vertical error of the straight movement detected by QPD without feedback control of laser, it can be known as $0.034 \mu\text{m}$; curve 'b' is the shape error of the flat measured by circular current sensor. Fig. 6 is the result of using laser datum, it can be seen from its curve 'b' that the method of laser datum can compensate for the movement error, and now the measuring is not effected by the movement error. The curve 'b' shows that the shape error of the flat is $0.14 \mu\text{m}/40 \text{ mm}$, its resolution is $0.02 \mu\text{m}$.

5. Conclusion

We have developed the laser datum system and performed the system behaviour test. Some conclusion remark can be drawn :

Using stability controlling system of the laser beam and adopting vacuum cover in optical path, we can obtain a high stability of 10^{-7} rad of laser beam. The laser beam is an ideal physical datum;

Using laser as straight movement datum, QPD as detector and piezocrystal as micro-driving structure, and adopting zero-position feedback controlling principle, we can get $0.1 \mu\text{m}/100 \text{ mm}$ straight movement accuracy;

There is no effect of the geometric movement for the measurement , the method of laser datum can attain $0.1 \mu\text{m} / 100 \text{ mm}$ or more high shape error measuring accuracy.

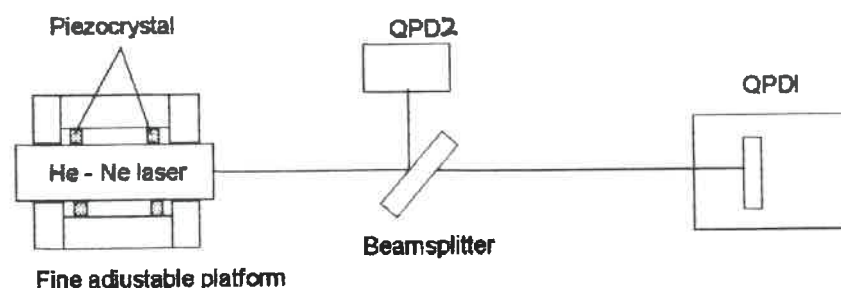


Fig. 1 Illustration of the stability controlling principle of the laser source

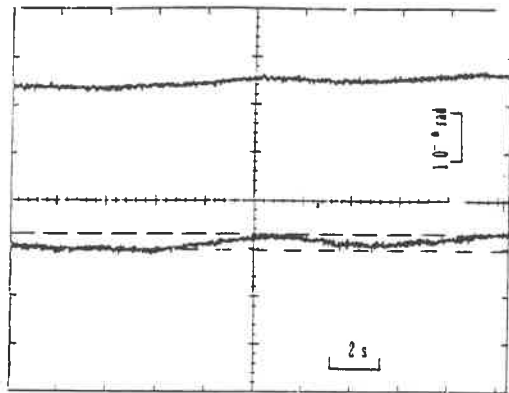


Fig. 2 Illustration of the beam stability without controlling of the laser source.

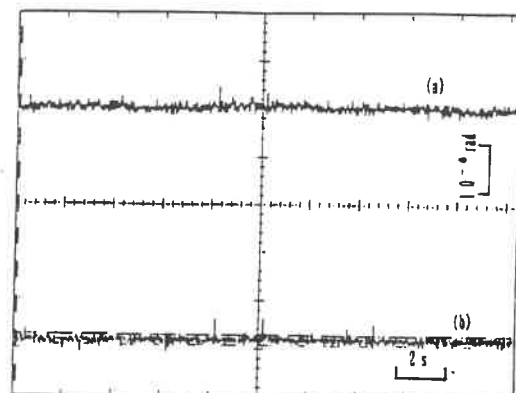


Fig. 3 Illustration of the beam stability with controlling of the laser source.

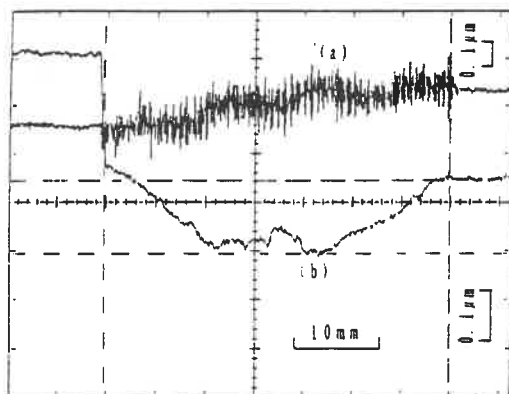


Fig. 4 Shape error measuring of optical flat without using laser datum

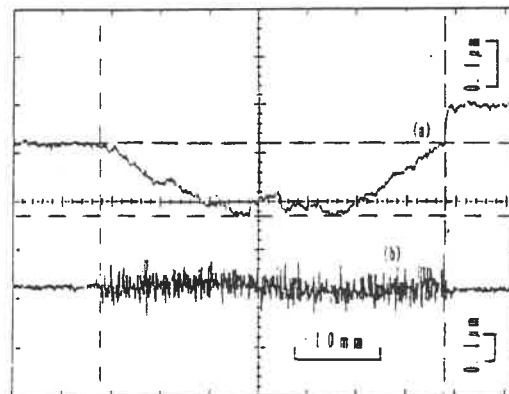


Fig. 5 Shape error measuring of optical flat of using laser datum

References

1. Hideki Matsumoto, et al, "An Ultra Precision Straight Moving system(2nd Report)—five Degrees Freedom Moving Control", JSPE-54-10, p1945, (1988).
2. Naoya Ikawa, et al, CIRP, 137(1), p523, (1988).
3. Wan De'An, et al, "Basic Experiemntal Research into laser Beam as a Positional Datum", IMCC, C80, (1991).

ZONE-PLATE INTERFEROMETER TO MEASURE THE POSITIONING ERROR OF A CUTTING TOOL

Takashi Nomura¹, Kazuhide Kamiya¹, Hiroshi Miyashiro¹,
Hatsuzo Tashiro², and Kazuo Yoshikawa²

1. Toyama Prefectural University, Toyama, 939-03, Japan

2. Toyama University, Toyama, 930, Japan

Spherical and aspherical mirrors are manufactured with an ultraprecision lathe. When the cutting tool of the lathe has a positioning error, the mirrors cut by the tool will have shape errors. To manufacture mirrors with accurate shape, the positioning error of the cutting tool has to be minimized.

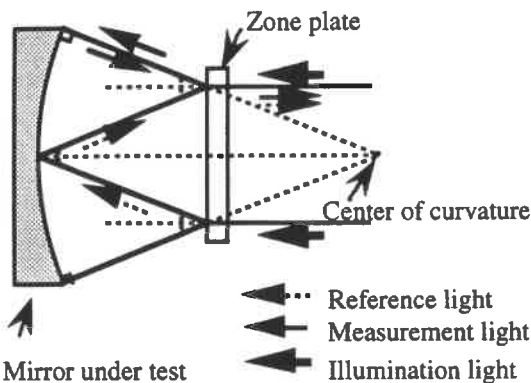


Fig. 1. Lights diffracted by a zone-plate

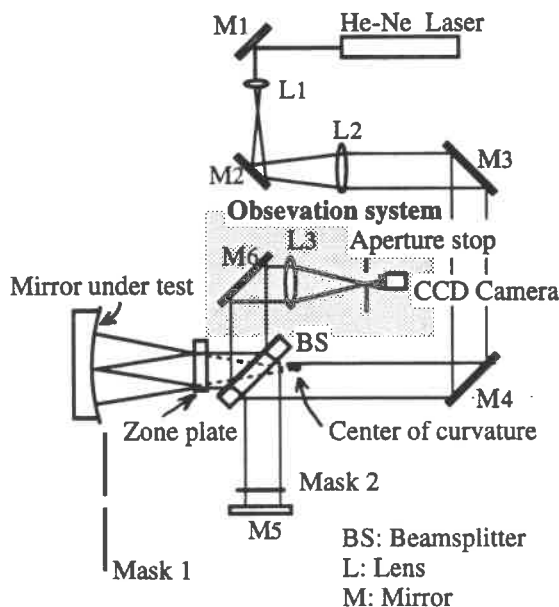


Fig. 2. Scheme of zone-plate interferometer

A simple type of zone-plate interferometer is developed to measure precisely the positioning error of the cutting tool. The construction of the interferometer is simple, because a collimation light is used as an illumination light and the shape of mirrors being tested is spherical. The special features of this interferometer are the location of a zone plate and the combinations of diffraction lights. When the zone plate is illuminated by a collimating light as shown in Fig.1, it diffracts converging and diverging lights. The zone plate is set at the mid-position between the center of the spherical mirror under test and the center of the curvature of the mirror. The zone plate is made in such a way that one of the first order diffraction lights goes perpendicularly to the mirror surface and another diffraction light goes to the center of the mirror surface. The diverging diffraction light is reflected on the entire surface of the mirror. The reflected light is divergently diffracted at the zone plate and the diffracted light goes to a charge coupled device (CCD) camera as a test light. On the other hand, the converging diffraction light is reflected at the center of the mirror surface. The reflected light is convergently diffracted at the zone plate and the diffracted light goes to the CCD cam-

era as a reference light. Figure 2 shows the scheme of the zone-plate interferometer. A spherical concave mirror is manufactured with a tool having a slight positioning error. When the mirror is measured with the interferometer, the shape error of the mirror and the positioning error of the tool can be obtained by analyzing the interference fringes.

To define the moving direction of the tool, Cartesian coordinate system is used. The axis of the spindle of the lathe is called z and the axes perpendicular to z are called x and y , as shown in Fig. 3. The cutting tool is moved in x - z plane from the minus side on the x axis to the center of the mirror

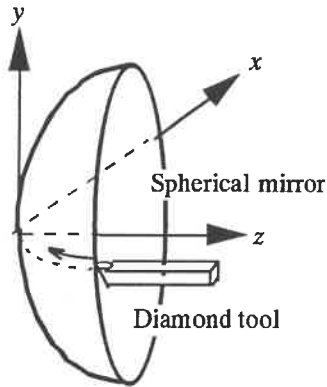


Fig. 3. Location of a tool indicated by Cartesian coordinate system

surface. We assume that the feed error of the tool is negligible. The positioning error of the z direction causes an error of thickness of the mirror, but does not cause a shape error of the mirror surface. The positioning error of the y direction causes a shape error near the central portion of the mirror surface. The positioning error is determined by observing near the central portion of the mirror surface with a microscope and is removed easily. On the other hand, the positioning error of the x direction influences the entire surface of the mirror. The error of the x direction is, therefore, discussed in this paper.

We suppose that the designed spherical mirror and the manufactured mirror are illuminated by a laser light diverging from the center of the curvature of the mirror. When the reference wavefront reflected on the designed spherical mirror and the wavefront reflected on the manufactured mirror produce interference fringes on the mirror surface, concentric fringes with an equal interval are observed. When a positioning error e of the x direction is sufficiently smaller than a curvature radius R and an aperture radius r of the mirror, the error e satisfied the following equation are obtained by calculating the optical path length,

$$e = r - \sqrt{R^2 - \left\{ \frac{nR\lambda}{2\sqrt{R^2 - r^2}} + \sqrt{R^2 - r^2} \right\}^2} \quad (1)$$

where n is the maximum order of the fringes, and λ is the wavelength of a light source. When r is sufficiently smaller than R , and λ is sufficiently smaller than r , the above equation is approximately expressed as follows:

$$e = \frac{nR\lambda}{2r} \quad (2)$$

The absolute value of the error is calculated by the equation. The sign of the positioning error is determined by the following method: The mirror under test are moved to the minus side of the x axis or the interferometer is moved to the plus side until the interference fringes without the carrier

component are observed on the half of the mirror along the x axis. The fringes obtained by the method are lateral-shearing interference fringes between the reference spherical surface and the tested surface. When fringes without the carrier component are observed on the right side of the mirror, the sign of the error is minus. On the contrary, when fringes without the carrier component are observed on the left side of the mirror, the sign of the error is plus. The moved distance of the mirror is equal to the absolute value of the positioning error.

The zone plate should be set at the mid-position between the center of the mirror under test and the center of the curvature of the mirror. In other words, the mirror should be precisely set at the designed position. To determine the position, reference flat mirror M5 is used. The collimation light reflected on beam splitter BS goes to mirror M5, reflected on it, and goes to the CCD camera. When mask 1 with a hole is set in front of the mirror under test, the light reflected at the reference mirror and the light reflected at the center of the mirror under test make interference fringes on the image plane of the CCD camera. The mirror under test is adjusted until null interference fringes appear so that the mirror is set at the correct position. When the mirror is measured, mask 1 in front of the mirror is removed and mask 2 is inserted in front of reference flat mirror M5. When the correct mirror is measured with the zone-plate interferometer, there are no interference fringes.

To test the validity of the method, a zone-plate interferometer was manufactured and three spherical concave mirrors were measured. The photograph of the zone-plate interferometer is shown in Fig. 4. Test mirrors Nos. 1 and 2 had been manufactured with the tools with different degrees of positioning error. Test mirror No. 3, an example of a spherical mirror without positioning error, had been manufactured not by cutting but by polishing. Figure 5 shows the image of the interference fringe pattern produced by the wavefront from the center of the test mirror No. 1 and the wavefront from mirror M5. The image indicates that the mirror under test was precisely set at the designed position. The positioning errors of test mirrors Nos. 1 and 2 are observed as distortions of interference fringes shown in Figs. 6 and 7. The experimental results show that these positioning errors are $-27\text{ }\mu\text{m}$ and $-3.5\text{ }\mu\text{m}$, respectively. The images obtained by the computer

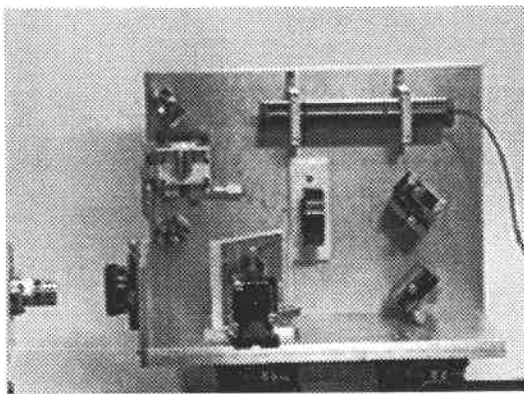


Fig. 4. Photograph of zone-plate interferometer

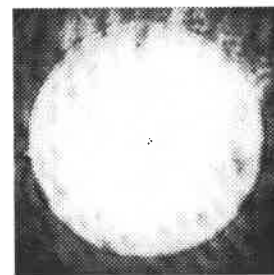


Fig. 5. Interference fringe pattern produced by the wavefront from the center of the test mirror No. 1 and the wavefront from mirror M5

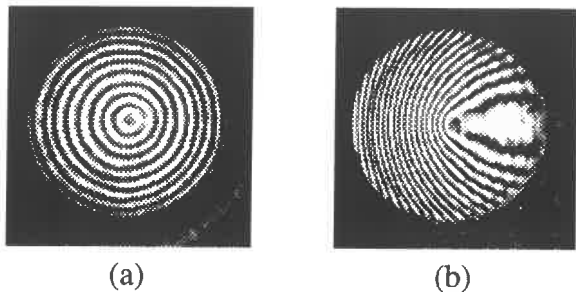


Fig. 6. Experimental results of fringe patterns of the test mirror No. 1. The mirror was cut by the tool with some positioning error

(a) Fringe pattern without a carrier component and (b) Fringe pattern with a carrier component at the half of the mirror on the x axis. The test mirror No. 1 was moved $-27 \mu\text{m}$ to the x direction.

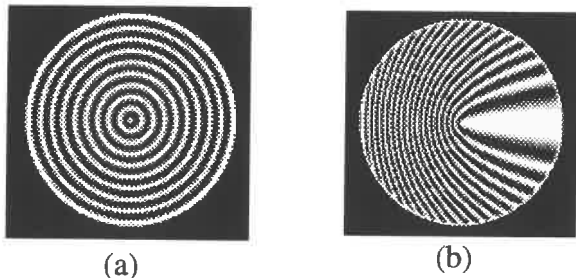


Fig. 8. Fringe patterns obtained by the computer simulation. The positioning error of the cutting tool is $-27 \mu\text{m}$.

(a) Fringe pattern without a carrier component and (b) Fringe pattern with a carrier component.

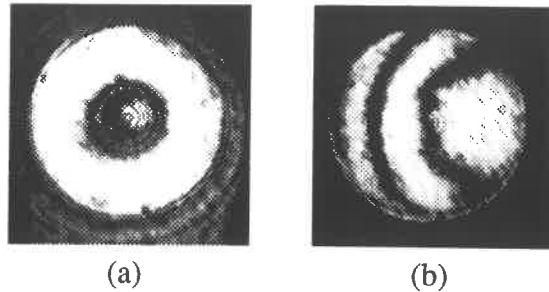


Fig. 7. Experimental results of fringe patterns of the test mirror No. 2. The mirror was cut by the tool with a slight positioning error

(a) Fringe pattern without a carrier component and (b) Fringe pattern with a carrier component at the half of the mirror on the x axis. The test mirror No. 2 was moved $-3.5 \mu\text{m}$ to the x direction.

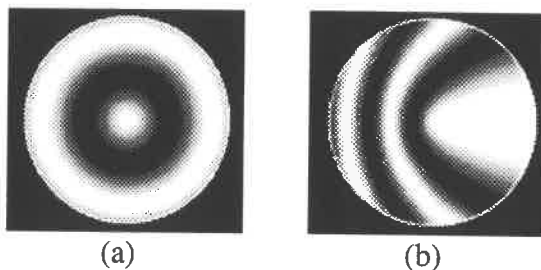


Fig. 9. Fringe patterns obtained by the computer simulation. The positioning error of the cutting tool is $-3.5 \mu\text{m}$.

(a) Fringe pattern without a carrier component and (b) Fringe pattern with a carrier component.

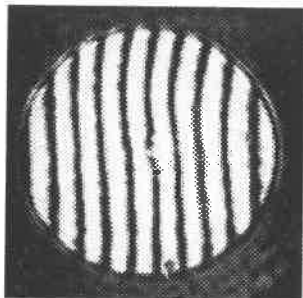


Fig. 10. Experimental result of a fringe pattern of the test mirror No. 3. The mirror was manufactured by polishing.

simulations, which were performed by the ray tracing method under the same conditions of the experiments, are shown in Figs. 8 and 9. The images obtained by the experiments agree well with the images obtained by the computer simulations. Figure 10 shows the fringes of the test mirror No. 3. The result showed that the interference fringes had no characteristic distortion caused by the positioning error. The measurement error of the proposed method is less than $0.9 \mu\text{m}$.

KEYWORDS: zone-plate interferometer, shape measurement of spherical mirror, positioning error of a cutting tool, on-machine measurement, computer generated hologram

Study of New Type Interferometer of Double Interferogram

Hongxi Zhu Boyin Lu Shenghua Ye

Precision Instrument Engineering Department, Tianjin University,
Tianjin 300072, P.R. China

Introduction

This paper introduces a new type of interferometric measuring system. Using the two reference mirrors abreast, this system combines two sets of interferometric fringes into one viewing field. The relation of the two sets of interferometric fringes changed based on certain rules due to the change of optical path length in the measuring system. With the aid of the photoelectric conversion and graphical analysis technology, it can be used in length and micro-angle measurement. The feasible experiments, the measured data and the conclusion from analysis are also introduced in this paper.

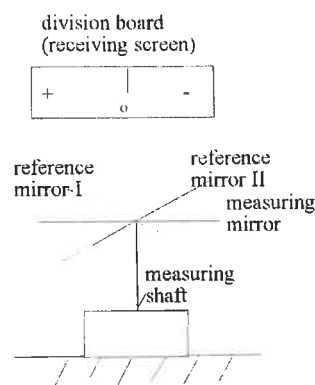


Fig. 1

Measuring principle of new di-reference mirror interferometer

The inclination angle of reference mirrors in optical path determines the density of interferometric fringes. The inclination direction (showed I and II in Fig.1) determines its $\pm$. (When the measuring mirror getting higher, the size getting bigger) When reference mirrors are in the direction I, the interferometric belt moves left, otherwise moves right.

If two reference mirrors are used abreast in interferometric optical path, we will get two sets of interferometric fringes with following rules and characteristics:

1. Length measuring principle of incoincident image

In the optical path, adjusting the clination angles of two reference mirrors to measuring mirrors to be both α , but in opposite directions, showed in Fig.2, there will be two sets of

interferometric fringes with interval $e = \frac{\lambda}{2n\alpha}$. When measuring mirror moves up and down, fringes will move in opposite direction. Every $\frac{\lambda}{2}$ the measuring mirror moves, every fringe interval the interferometric fringe moves. From the beginning state of fringes overlapping, the two sets of interferometric fringes relatively move a half interval $\frac{e}{2}$, then the fringes will overlap once. If optical path length is changed from the beginning state of fringes

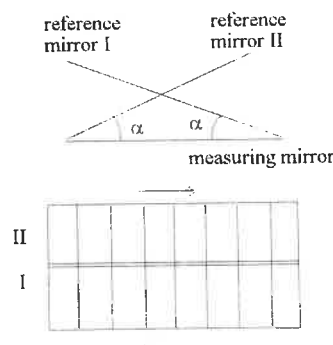


Fig. 2

overlapping, fringes overlap N times, optical path length will change $N \cdot \frac{\lambda}{4}$. The sensitivity of length measurement in this method enhances one time to the ordinary method.

2.Length measuring principle of vernier

Adjusting clination angles of two reference mirrors to measuring mirrors to be α and β respectively ($\alpha \neq \beta$), but in opposite direction. Then two sets of interferometer fringes will be with intervals e' and e . If it can be fit with $ne'=(n-1)e$, showed in Fig.3, adjusting measuring mirror to make three pairs of interferometric fringes A_0B_0 , A_nB_{n-1} , $A_{-n}B_{-(n-1)}$ in two sets of interferometric fringes align simultaneously. The overlapping state of three pairs of fringes is destroyed by moving measuring mirror in obverse direction. When

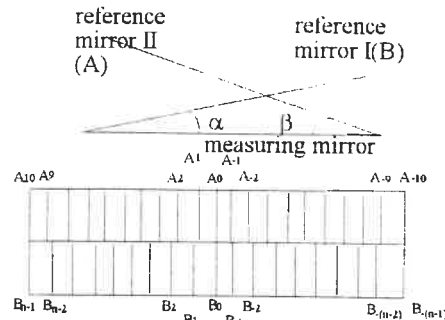


Fig.3

moving $\frac{1}{n} \cdot \frac{\lambda}{4}$, A_1B_1 and $A_{-n}B_{-(n-1)}$ overlap, when

moving $\frac{2}{n} \cdot \frac{\lambda}{4}$, A_2B_2 and $A_{-(n-1)}B_{-(n-2)}$ overlap, and so on and so forth. When moving

$\frac{n}{n} \cdot \frac{\lambda}{4}$ three pairs of fringes of $A_nB_n, A_{-1}B_0, A_{-(n+1)}B_{-(n-1)}$ are simultaneously overlap again. After three pairs of fringes aligned, overlap M times, and the K th pairs of fringes overlap, the

moving distance of the measuring mirror is $L = (M + K \cdot \frac{1}{n}) \cdot \frac{\lambda}{4}$

If $n=10$, the sensitivity can reach to $\frac{\lambda}{4n} = 0.015\mu\text{m}$.

3.the principle of angular measurement

As the former stated, adjusting both angles between reference mirrors and measuring mirrors to β , but in opposite direction, the intervals of two sets of interferometric fringes are equal, when the measuring mirror moves α related to the reference mirrors, (showed in Fig.4), the angle of the measuring mirror to the reference mirror 1 is $(\beta - \alpha)$, the angle to the reference mirror 2 is $(\beta + \alpha)$, and the intervals of the two sets of interferometric fringes respectively are:

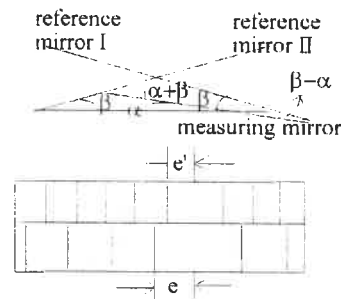


Fig.4

$$e = \frac{\lambda}{2n(\beta - \alpha)} \quad (1)$$

$$e' = \frac{\lambda}{2n(\beta + \alpha)} \quad (2)$$

take $e = ke'$, apply (1), (2) to it.

That is $\alpha = \frac{k-1}{k+1}\beta$, β is a fixed number. Based on angle β and the changes of the interferometric fringes intervals, the micro-angle can be determined.

Feasible experiment

The above analysis is correct theoretically. In order to textify its feasibility, resolving experiments have been done on vertical contact interferometer. The practical measurement has been done with simple experimental equipment. The specific experimental method and measuring data are as follows:

1. Resolving experiment on the vertical contact interferometer

Showed as the measuring process frame in Fig.5, based on the Coincident image and vernier length measuring principle, compared the slight moving working plat with laser interferometer or inductance micrometer to be the criterion, to make the contact interferometer with the reference mirrors in the observe and reverse states, we can do one group of measurement separately, then analyse and compare the measured data and theoretical data. A part of the interferometric length measuring data are listed in Table 1.

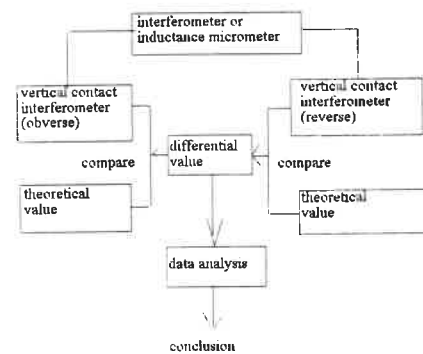


Fig.5

Table 1 Incoincident Image Measuring Data

unit: μm

times	contact interferometer data		theoretical data	differential value (measurement-theory)		differential value of obverse and reverse state
	obverse	reverse		obverse	reverse	
1	0.15	0.13	0.14	0.01	-0.01	0.02
2	0.29	0.25	0.28	0.01	-0.03	0.04
3	0.43	0.38	0.42	0.01	-0.04	0.05
4	0.57	0.53	0.57	0	-0.04	0.04
5	0.71	0.70	0.71	0	-0.01	0.01

The maximal difference between theoretical data and measuring data in obverse state is , $0.01\mu m$,that in reverse state is $0.04\mu m$. The different value in obverse and reverse state is $0.05\mu m$.

2.Measurement with simple experimental equipment

Using the simple measuring equipment showed in Fig.6, two sets of interferometric fringes got from interferometric system can be combined into one viewing field directly. It can be converted into the measuring data listed in Table 2 through the combined position information from photoelectric conversion.

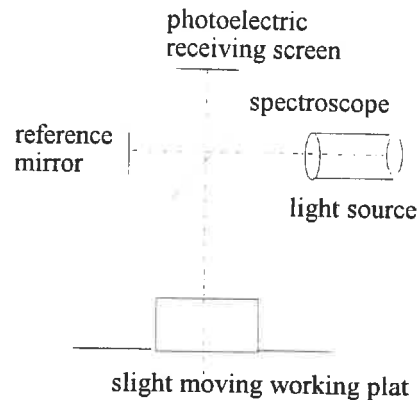


Fig.6

Table 2 Coincident Image Measuring Data in Simple Measuring Device unit: μm

times	one		two		three		four		five	
	data	differen- tial value	data	differen- tial value	data	differen- tial value	data	differen- tial value	data	differen- tial value
1	0.15	0.15	0.12	0.12	0.20	0.20	0.20	0.20	0.20	0.20
2	0.32	0.17	0.29	0.17	0.31	0.11	0.39	0.19	0.39	0.19
3	0.52	0.20	0.43	0.14	0.50	0.19	0.52	0.13	0.55	0.16
4	0.68	0.16	0.59	0.16	0.70	0.20	0.64	0.12	0.71	0.16
5	0.82	0.14	0.74	0.15	0.87	0.17	0.81	0.17	0.87	0.16

$$\lambda = 0.6328\mu m, \text{ measuring length } L = N \cdot \frac{\lambda}{4}$$

Theoretical length value is $\lambda/4 = 0.158\mu m$, when the fringes overlap one time.

Conclusion

Theoretical analysis and feasible experiment show that some rules and characteristics of the new type of di-reference mirrors can be used in actual geometric sense measurement. It is worth studying and developing.

Computer Vision System for 2-D Edge Detection Using Hardware Demodulation

Jing Fangsheng , Song Jin , Li Dongguang , Ren Ying
Precision Instrument Engineering Department ,Tianjin University
Tianjin 300072, P . R . China

Introduction

A computer vision system commonly consists of a solid state CCD camera as a sensor that has a format of matrix pixels and an interface protocol that connects the camera with vision system .This is an analogue signal protocol , so the intrinsically discrete data in camera must be converted to an analogue signal and sent to the vision system computer where it is digitized into a 480 by 512 matrix of intensity values between 0 and 255 , as seen by the computer .The edge detection consists of two steps, a filtering step and a differentiation step. Edge detectors are mathematical operations. In order to assign reliably the edge to a precision higher than one pixel, the edge finder is fairly sophisticated. [1]. In some cases of dimensional measurement (such as diameter of small shaft, diameter of small ball and form of cutting tool) , the objective plane intensity of light can be expressed as function :

$$I(x) = I_0 * \left[1 - \text{rect}\left(\frac{x}{2A}\right) \right] \quad (1)$$

where : $2A$ is the dimension between two edges ;
 I_0 is maximum intensity of light .

The image in the camera field of view has dark-to-light transition of a few pixels. Zhou Zhongli has theoretically analyzed this specified condition[2]. According to his analysis, the ideal demodulation network can be replaced by a low-pass filter for high contrast edge .

The vision system described in this paper consists of a hardware video signal processing circuit that is designed for high contrast edge to realize the filtering and differentiation steps . So the job of data collection and data processing is fairly simple. Performance tests are presented which show that the accuracy of the 2-D edge determination of the system is about $2\mu\text{m}$ (3σ) and the measuring cycle is less than one second .

Edge detection using hardware demodulation

The edge detection circuit must solve three problems:

- (a) To restore sampling pulses to continuous electrical signal .
- (b) To determine the critical points as edge points from this continuous electrical signal .
- (c) To match with the PC in time-sequence .

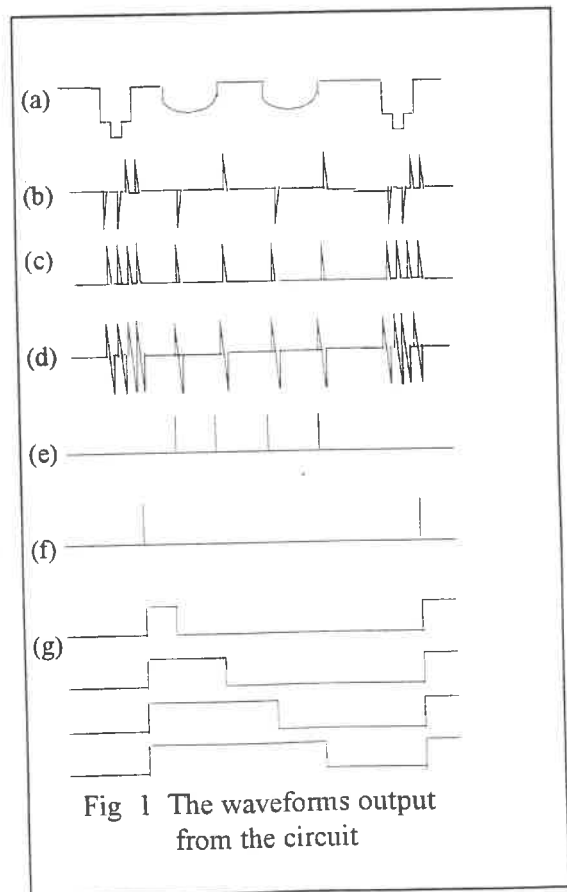


Fig 1 The waveforms output from the circuit

A low-pass filter that has linear phase character and about 10^6 rad/s cut-off frequency is used for demodulation of the video signal. The waveform of the restored signal is shown in Fig. 1 (a) . The waveforms which are generated by once differentiation circuit , absolution circuit and twice differentiation circuit are shown in Fig. 1(b) , Fig. 1(c) and Fig. 1(d) separately. The first four zero-crossing points of the twice differentiation waveform correspond to leading side of line-blanking, leading side of line-sync, trailing side of line-sync and trailing side of line-blanking in Fig.1(a) respectively, and the others correspond to the edge points of measured object . Through the zero comparator and the pulse selection logic circuit, the edge point pulses are generated and selected, as shown in Fig. 1(e). The trailing side of line-blanking is chosen as start point of each line, as shown in Fig. 1(f) . Fig. 1(g) shows the output from the square forming circuit that gives square waves from the start point to edge points. Through inserting high frequency stable pulses and counting the pulse numbers by counters, the x-coordinate of edge points in the view field can be recorded by counters. The clear signal of counters must be ahead of the start point, so the trailing side of line-sync is used for clearing counters. The leading side of line-blanking is used for interrupt request to PC and the interrupt service routine does the job of data collection . It should have enough interval from "interrupt requesting" to "clearing counters", so that the service routine of data collection can be completed reliably. Because of the reason above, the duration of line scan is increased by reducing the frequency of CCD scan signal.

The vertical sync signal is used to request another interrupt of PC and the corresponding interrupt service routine gives the beginning mark of each field by writing "FFFF"to the data memory. In the main program, the collected data are rearranged. According to the beginning mark and following data in the data memory, the data numbers and the line numbers of each field are recognized. Whether even field or odd field can be judged by the line numbers of the field. Through linking the data of two adjacent fields, the entire data of a frame is obtained.

Experiments

Setting scale

There are two methods which can be used for setting scale:

- 1) A gage object of known dimension, usually a wire gage, is focused at the field of view and the diameter of the wire gage is measured using linear fits to short segments of the two edges of the wire. The diameter, in pulse, is used with the known diameter of the wire gage to set the scale of the field of view in $\mu\text{m} / \text{pulse}$. The experiment has shown that the scaling figure could be reduced by using a larger diameter wire gage. In other words, this method brings about an uncertainty in setting scale. The following shows that this is caused by the demodulation error.
- 2) The right method for setting scale is using a wire gage that can be moved precisely and parallel. The moving distances of the same side of the wire gage are used as standards in setting scale.

A tool measuring microscope, whose eyepiece is replaced by a video camera , is used for demonstrating the scaling and measuring processes .

A wire gage is mounted on the working table of the microscope in the direction making an angle of 90° with scan lines in the field of view. A set of block gages of 1.0mm, 1.2mm, 1.4mm 1.6mm and 1.8mm is used to give accurate linear displacements of 0 mm, 0.2mm, 0.4mm, 0.6mm, 0.8mm in the x-direction, which is parallel to the scan lines of the video camera. By using linear fits

Table 1: Measured data of setting scale in x

data	j1	j2	j3	j4	j5
i1	3001	3798	4591	5386	6181
i2	2998	3797	4589	5384	6181
i3	2999	3796	4592	5384	6183
i4	3001	3797	4589	5385	6184
i5	2998	3796	4592	5387	6182

to short segments, five segment readings are taken at each location and the segment coordinates, in pulse, are recorded in Table 1. The standard displacement between neighboring positions in x-direction is $200\mu\text{m}$, so the scaling value in x-direction can be calculated:

$$X_s = \frac{200}{\left(\sum_{i=1}^5 \sum_{j=2}^5 (d_{i,j} - d_{i,j-1}) \right) / 20} = 0.2513 \mu\text{m/pulse} \quad (2)$$

Where $d_{i,j}$: a value in i-row and j-column of Table 1; X_s : pulse equivalent.

The equivalent Y_s , the distance between arbitrary two neighboring scan lines in the view field, can be determined by rotating the working table through a standard 45° with the help of a 45° standard angle block gage and an optical-electrical autocollimator. The distance between middle points of two neighboring segments of the wire gage in x direction is equal to $n*Y_s$, where n is the number of scan lines passes through the segment which equals to 50 in the experiment, so the equivalent can be obtained:

$$Y_s = \frac{X_s * \left(\sum_{i=1}^4 \sum_{j=1}^5 (d_{i+1,j} - d_{i,j}) \right)}{20 * 50} = 2.655 \mu\text{m/line} \quad (3)$$

where: $d_{i,j}$ is taken by the same method as above.

Demodulation error

A number of experiments have shown that there are systematic errors caused by the demodulation circuit of the vision system. The systematic errors are called demodulation errors, and the demodulation errors can be obtained by measuring a standard metal ball.

There are two cases for the measured edges in the view field when the time sequence of scan lines is taken into consideration: light-to-dark edges and dark-to-light edges. The demodulation errors are mainly generated in the last case.

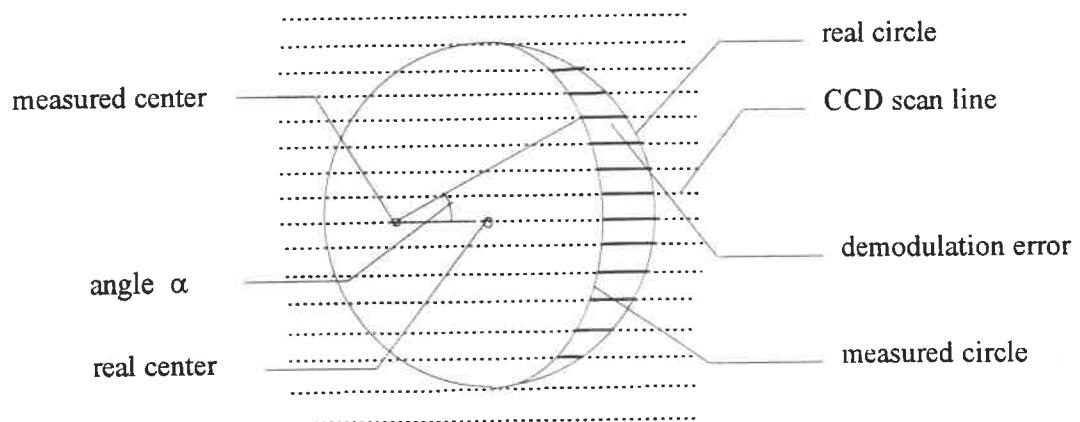


Fig 2 Demodulation error in the measured metal ball

A metal ball with a diameter of 1mm is used for finding the demodulation errors. Fig. 2 shows the measurement results: The left half of the circle that presents light-to-dark edge of the ball has negligible errors when comparing the measured radius with the real radius. The right half of the circle that presents dark-to-light edge of the ball has demodulation errors in x-direction that depend on the angle α which is between edge's normal line and scan line. The errors are shown in Table 2.

Table 2 : Demodulation error

angle ($\pm \alpha^\circ$)	error (μm)	angle ($\pm \alpha^\circ$)	error (μm)	angle ($\pm \alpha^\circ$)	error (μm)	angle ($\pm \alpha^\circ$)	error (μm)
0----24	-30	53----55	-23	64----65	-16	71----72	-8
24----33	-29	55----57	-22	65----66	-15	72----73	-6
33----39	-28	57----59	-21	66----67	-14	73----74	-4
39----43	-27	59----61	-20	67----68	-13	74----75	-2
43----47	-26	61----62	-19	68----69	-12	75----76	0
47----50	-25	62----63	-18	69----70	-11	76----90	0
50----53	-24	63----64	-17	70----71	-10		

Some measurement results

The following, shown in Table 3, gives the measurement data of a set of wire gages that are placed at right angle with scan lines in the field of view. The measurement deviations are $-30 \pm 1\mu\text{m}$. If a compensating value of $30\mu\text{m}$ predicted by Table 2 is added to the measurement data, the measurement accuracy is about $1\mu\text{m}$ (3σ).

Table 3: Measurement data of a set of wire gages (unit: mm)

real diameter	measured diameter	deviation	real diameter	measured diameter	deviation
0.170	0.140	-0.030	0.232	0.203	-0.029
0.291	0.262	-0.029	0.402	0.372	-0.030
0.461	0.430	-0.031	0.724	0.693	-0.031
1.008	0.977	-0.031	1.441	1.411	-0.030

In order to demonstrate 2-D edge measurement, a set of metal balls is measured. The left and right halves of the circle are calculated separately by means of the least square fits. From the calculation results, we can find the deviations of left radius and right radius are $\pm 1\mu\text{m}$ and $9.5 \pm 1.5\mu\text{m}$ respectively, the distances between left center and right center are $40 \pm 5\mu\text{m}$. Through compensating the data, using Table 2, the measurement error of radius is $2\mu\text{m}$ (3σ) and the position error of center is $1.5\mu\text{m}$ (3σ).

Conclusion :

In the paper, a computer vision system for 2-D edge detection using hardware demodulation has been described. This system is only used for the case of high contrast edges in the view field. The filtering step and differentiation step are implemented by using the hardware circuit. This demodulation circuit simplifies the edge finder and reduces the measuring cycle.

The method for setting scales is presented. By means of the practical measurement of some standard workpieces, the demodulation errors are discovered and the error table is obtained. Through error compensation, the accuracy of edge determination of the system can achieve about $2\mu\text{m}$ (3σ).

References:

- (1) T. D. Doiron, Computer vision based station for tool setting and tool form measurement , Precision Engineering , Oct. 1989 , Vol. 11 No. 4
- (2) Zhou Zhongli ,The demodulation method of dimensional measurement using CCD sensor. Chinese Journal of Scientific Instrument , Feb. 1986 , Vol. 7 No. 2

A New Fringe-field Capacitive Sensor

Tian Guiyun

Yan Bin

**(Measurement & Control Engineering Department
Sichuan Union University, Chengdu, Sichuan 610065 /P. R. China)**

1. Abstract

A new capacitive sensor based on the principle of fringe-field is introduced in the paper. Because it bases on the fring-field , not on the parallel field, its structure can be designed flexibly to meet complicated requirement of measurement and it can effectively turn change of non-metallic into change of capacitance by non-contact measurement. Several representative structures are presented and relevant experimental results are given. Meanwhile, its application of non-contact measurement in chemical industry is discussed especially.

2. Introduction

The change of tested objects (such as elements of substance) can cause the change of capacitance by traditional capacitive sensor. Because of its high sensitivity, non-contact measurement, high dynamic frequency response and fine temperature stability, it is widely used in all kinds of field. Compared to the traditional capacitive sensor, the fringe-field capacitive sensor possesses not only these advantages, but also marked characteristics that flexible and small-sized structure, and particularly its high sensitivity to dielectric constant of non-metallic. That is to say, any slight variation which can result in variation of dielectric constant (such as variation of humidity, density, solution concentration, classification of material and so on) will lead to remarkable variation of capacitance, therefore we can measure and obtain the slight variation and get the high resolution. Because of these characteristics, it can be applied to chemical measuring widely.

3. Principle and discussions

According to electric field principle, field lines of capacitor are from the positive plate electrode to the negative plate electrode through air and object being tested. The capacitance generated is :

$$C=Q(S)/V=\frac{1}{V}\int \epsilon(S)\vec{E}(S)\cdot d\vec{S} \quad (1)$$

Where,

V--driver voltage

S---integral camber

$\epsilon(S)$ — dielectric constant

including the constant ϵ_1 of air space and ϵ_2 of tested object.

$\vec{E}(S)$ ---electric strength of any position along the integral camber.

both $\epsilon(S)$ and $\vec{E}(S)$ are the function of position.

$d\vec{S}$ --- unit vector along the direction of unit camber dS .

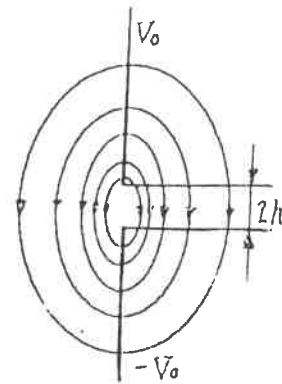


Fig. 1

We discuss the specific case of opposing semi-infinite plate separated by a distances $2h$ in a uniform dielectric (see Fig.1). Assuming that the plates have potential V_0 and $-V_0$ respectively, the electric stream function in the intervening space can be shown as this :

$$U(y)=\frac{2V_0}{\pi}\ln\left[\frac{y+\sqrt{y^2-h^2}}{h}\right]+iV_0 \quad (2)$$

Assuming the original semi-infinite plate is shortened to height H . Then if $H \gg h$, the field is approximately constant at the distance H from the origin. Therefore, calculating the total flux by evaluating the stream function at $y=H$ and $y=h$, we get the charge Q :

$$Q=2\epsilon\{U(H)-U(h)\}=\frac{4}{\pi}\epsilon V_0\ln\{[H+\sqrt{H^2-h^2}]/h\} \quad (3)$$

Assuming $H \gg h$, the effective capacitance, per unit width W , is :

$$C_s = \frac{Q}{2V_0} = \frac{2\epsilon}{\pi} W \ln\left(\frac{2H}{h}\right) \quad (4)$$

Following that we can discuss the sensitivity relevant to displacement and the dielectric constant. Depending on the result shown above (formula (4)), we get (1). Sensitivity relevant to displacement (other parameters are constant.).

$$K_h = dC_s / dh = - \frac{2\epsilon W}{\pi h_0} \quad (5)$$

(h_0 is the primitive distance .)

In this case, the sensitivity is in direct proportion to ϵW , in inverse proportion to h_0 and irrelevant to H . In an other word, if width W is small, we can obtain high sensitivity by adjusting primitive h_0 . Now we can draw an important conclusion that the sensor may be small-sized structure.

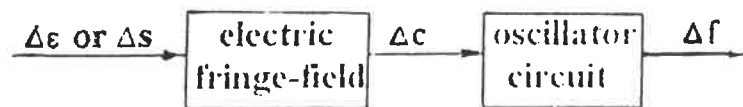
(2). Sensitivity relevant to dielectric constant (other parameters are constant.)

$$K_e = dC_s / d\epsilon = \frac{2}{\pi} W \ln(2H/h) \quad (6)$$

Due to $H \gg h$, the sensitivity is remarkable even if width W is small. Our researching emphasis is on various factors which affect dielectric constant ϵ (including the dielectric constant of air space and dielectric constant of material being tested). Based on this deduction, we have designed several representative structures to measure non-metallic objects, including the defect, uniformity, elements, classification of material, moisture, humidity, and concentration of solution.

4. Experiments

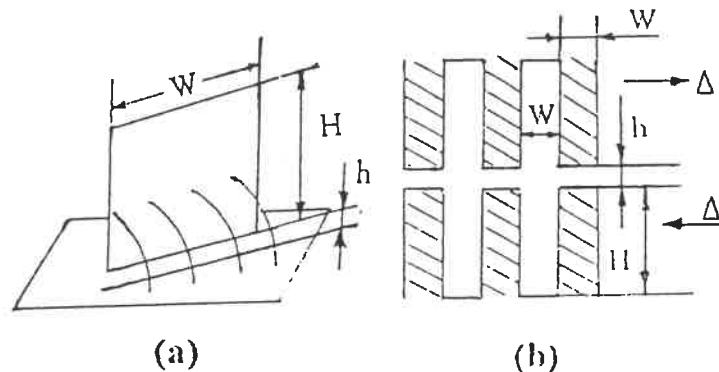
(1). Depending on these discussions, we design the transfer current for the fringe-field capacitance sensors. Figure 2 illustrates a hardware configuration.



$\Delta\epsilon$ --- variation of dielectric constant Δs --- variation of displacement.
 Δc --- variation of capacitance Δf --- variation of frequency

Fig. 2

(2). Guided by the deduction above discusses, fringe-fields for different electrode plate structures are designed to research the subject of non-contact measurement on dielectric constant ϵ of non-metal and on metal. Fig. 3 show s different structures and their curves of fringe-field that we have made the experiments. Other structures can be designed according to requirements of measurement.



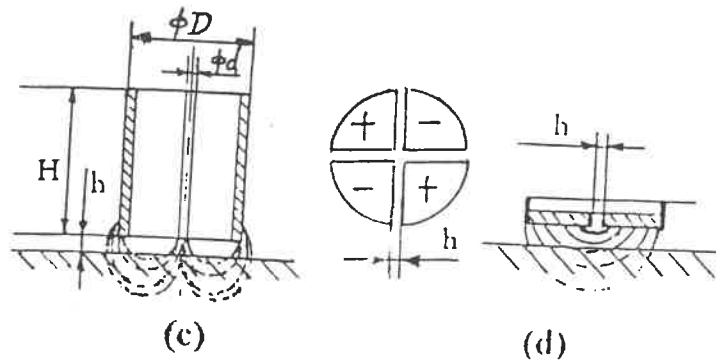


Fig.3

Note:

- (a). In these experiments, thickness of electrode is less than $5 \mu\text{m}$.
- (b). $h \ll H$, and $h < 2 \text{ mm}$.
- (c). Fringe-field shown in Fig. 3(a) is used to measure the displacement of metal. Others is used to measure non-metallic material.
- (d). Tested objects include the defect, uniformity, elements, classification of material, moisture, humidity, and concentration of solution.

5. Conclusions

After experimenting, we can draw an conclusion that if the primitive distance (h_0) is chosen suitably, the sensitivity of fringe-field sensor is high enough to respond slight change of tested object. The stability and repeatability is satisfying with precision engineering and analysis researches. The sensor can be used for high accurate non-contact measurement, especially for non-metallic materials. Because of these characters, the sensor based on fringe-field can solve the problems of non-contact precision measurement for which it is difficult to use photoelectric sensors, such as measuring ceramics optical glass and high performance compound materials(such as diamond film, leather and so on).

Reference

1. J.L. Garbini, R.A. Saunders, and J.E. Jorgensen. In-process drilled hole inspection for aerospace application, *Precision Engineering*, 1991 (2) .

FIBRE OPTIC SENSOR ON ROBOT END EFFECTOR FOR FLEXIBLE ASSEMBLY

*K.L. Yung, W.S. Lau, C.K. Choi, Y.Y. Shan
Department of Manufacturing Engineering
The Hong Kong Polytechnic University
Hung Hom, Kowloon, Hong Kong
E-mail: mfklyung@hkpucc.polyu.edu.hk*

A fibre optic sensor system was constructed for use on robot end effectors for flexible assembly. The sensor detected the deviations between robot end effector and the workpiece. The signal was fed back to robot controller to shift the end effector until the centre of end effector and the centre of workpiece were aligned at the correct orientation. Then workpiece can be grasped symmetrically. Sensor fusion concept was used to guard against sensor system failure. Fuzzy linguistic variable and control rule concept were introduced in the sensor integration. The experimental setup for the sensor integrated system was shown. The accuracy was also discussed.

INTRODUCTION

Despite the general industrial environments are favorable for the application of robots in flexible assembly, there remain some unforeseeable causes and factors to generate errors in assembly operation. For example, when robot is ready to grasp workpieces, the robot end effector may have position and orientation deviations with respect to the workpiece. The deviation is caused by either the errors of mechanism in robot teaching and programming procedures or in the workpiece delivering procedure. Such pre-grasping error has seldom been considered in other research. Actually such errors affect not only the position of the grasp, but also the accuracy of the subsequent assembly operation. These errors may not be acceptable in some manufacturing procedures such as precision assembly and safety grasping of some dangerous parts. It is needed to detect certain errors before they give rise to assembly error so that the assembly process may proceed without interruption from some minor errors.

For the comparatively minor displacement between the workpiece and the

robot end effector, it is more reasonable to be corrected by sensors on the robot end effector. Due to the limited size of the end effector, the sensors should be small in size and light in weight, of non-contact type, and have analogue output. The reflective optical fibre sensors have all these characteristics, rendering it to become one of the more suitable choices for this type of application.

PRINCIPLES

The simplest method to correct the deviation of the end effector with respect to the workpiece is to mount a pair of optical fibre proximity sensors on both inner sides of the robot end effector. The signals from the two sensors send out to detect the distances from the inner sides of the end effector to both surfaces of the object to be grasped. The signal which represents the position deviation is fed back to the robot controller. Accordingly the robot controller regulates the position of the end effector to minimize the error. However this can only determine one dimension of deviation.

The two dimensional deviation can be resolved by mounting two pairs of fibre sensors on the end effector with each pair on both inner sides. The illustration is in

Fig.1. The outputs of these four sensors are processed with a pre-determined algorithm to generate the error signals that represent the deviations of the end effector with reference to the workpiece. According to these signals, the position of the end effector is controlled until the central reference points of end effector and the centre of workpiece are match with the correct orientation, as a result, the workpiece will be grasped symmetrically.

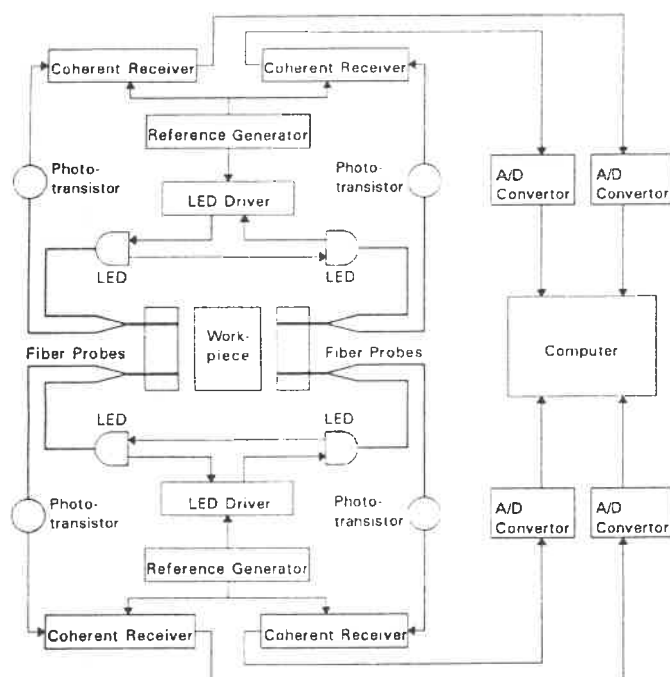


Fig. 1 Two dimensional deviation correction by mounting two pairs of optical fibre sensor to robot end effector and related pressing circuits

The possible relationships between sensors and rectangular and cylindrical workpieces are presented in Fig.2. The required correct positions are shown in Fig.2(a) and (e), in which all the detected signals are equal. For the deviations, following equations are used to generate the needed control signals for Fig.2(b), (c), (f), (g) and (d), (h)^[1]:

$$D_{rx} = \frac{1}{2}[(d_1 + d_3) - (d_2 + d_4)] \quad (1)$$

$$D_{ry} = \frac{1}{2}[(d_1 + d_4) - (d_2 + d_3)] \quad (2)$$

$$D_{cx} = \frac{1}{2}[(d_1 + d_3) - (d_2 + d_4)] \quad (3)$$

$$D_{cy} = \frac{1}{2}[(d_1 + d_2) - (d_3 + d_4)] \quad (4)$$

Since it is a kind of null detection method, the realization is convenient and reflectivity deviations can be neglected.

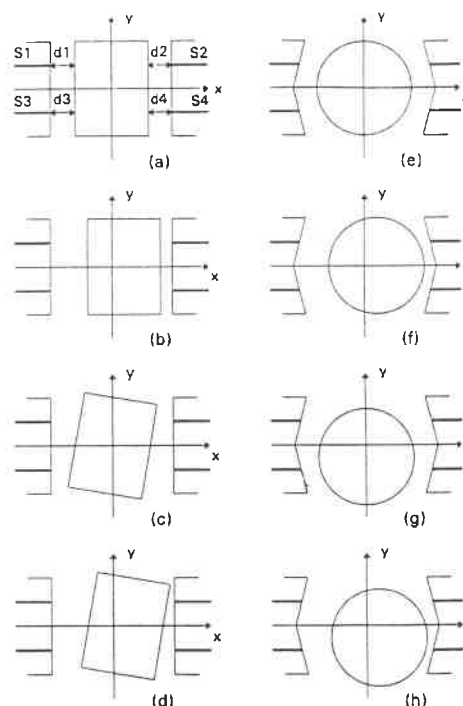


Fig. 2 Possible relations between sensors and the rectangular and cylindrical workpieces

In the above system, emphasis has been focused on the deviation correction of the robot end effector with respect to the

workpiece. The assumption is based on no faults occurred to the sensor itself. Actually, sensors will inevitably suffer an occasional loss of data or damages. The whole sensor system will be ruined only one of the sensor is malfunctioned. The problem lies in that no sensors or signals are used to monitor the status of the sensors. It means that the system can not find out whether a sensor signal is reliable or which sensor is faulty. We solve this problem by adding a small parallel movement, timeless and costless, to find the needed information. Besides the four detection signals as discussed above, another four redundant sensor signals are provided. Using the message of distances and angles of inclination, several criteria can be found out. Thus the sensor system can normally be operated even with one sensor having problem. This has shown that the system robustness is enhanced and the system possesses the abilities of self detection and protection.

Workpieces delivered on a conveyer may lie in incorrect positions or even fall down. By mounting two sensors at the bottom sides of the gripper to accompany the four sensors mentioned above, a detection system was formed to find out whether the workpiece has fallen down or having unacceptable displacement. Fuzzy linguistic variable and control rule were introduced to cope with the decision making and control process based on the sensor outputs for the situation. The change of state from bottom sensor detection to side sensor orientation correction was also determined by fuzzy control method.

EXPERIMENTAL SETUP

The system setup block diagram is as shown in Fig.3. It consists of a Sony SRX-4CH assemble robot and its controller, optical fibre sensor system, data acquisition

board PC-LPM-16, and computer.

Fig.4 illustrated the experimental setup for deviation correction. Fig.5 illustrated the experimental setup for six sensors consisting of a detection system.

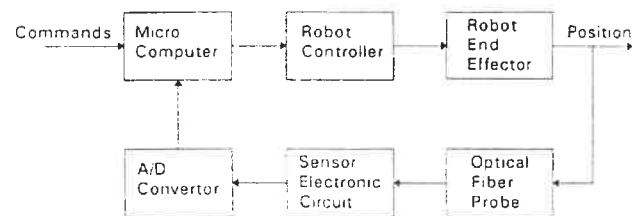


Fig. 3 System setup

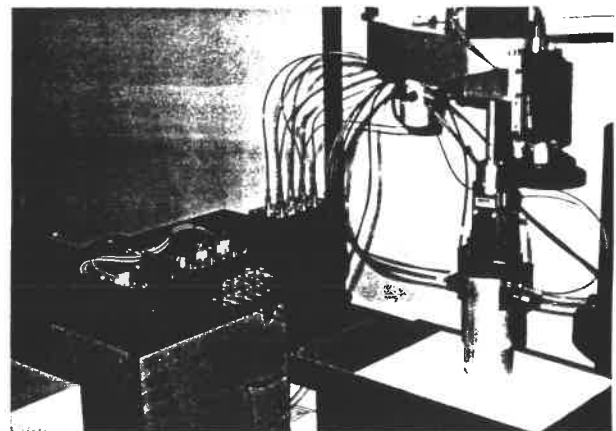


Fig. 4 Experimental setup for deviation correction

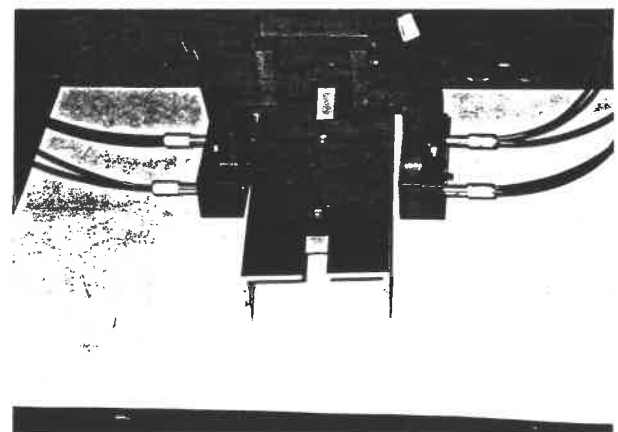


Fig. 5 Experimental setup for six sensors consisting of a detection system

ACCURACY DISCUSSION

The repeatability of the assembly robot

being used is specified as 0.05 mm in x, y and z axes, and $\pm 0.05^\circ$ in R axis^[2]. The method discussed utilizes the characteristics of the robot repeatability.

Through measurement we obtained the resolution of every channel of the fibre optic sensor to be reaching 0.005 mm through its dynamic range from 3 mm to 11 mm, which falls within the range for optical fibre sensor far distance usage^[3]. It can achieve even higher resolution in the near distance usage.

In the integration system, the resolution depends on the bits of A/D converter. The specifications in the PC-LPM-16 board are as follows: the least significant bit LSB of the 12-bit conversion value is equivalent to 2.44 mv in the 10 v (± 5 v) range; reading ranges from -2,048 to +2,047 if the board is configured for bipolar operation. In the experiment, the input voltage is set approximately from -1.0 v to +1.0 v, the correspond reading is from -400 to +400. Then the resolution is $(11-3)\text{mm}/800 = 0.01$ mm. Compared with the robot repeatability, 0.05 mm in x, y and z direction, the resolution is acceptable.

We can reasonably define the combined accuracy when robot system integrated with the sensor systems as follows:

$$\begin{aligned}\Delta_{\text{inte}} &= \sqrt{(\Delta_s)^2 + (\Delta_r)^2} \\ &= \sqrt{(0.01)^2 + (0.05)^2} \quad (5) \\ &= 0.051 \text{ mm}\end{aligned}$$

where Δ_s is the sensor resolution, Δ_r is the robot repeatability. It is shown that the sensors apparently contributed neglectable additional error to the integrated system.

Although the maximum speed of the robot is specified 5 m/sec, when gripper approaches the workpiece, it does not move at this maximum speed. We may calculate the approach speed in the following way. The maximum sample rate in PC-LMP-16 is

50,000 points/sec or $20\mu\text{s}/\text{point}$. To ensure the 0.01 mm resolution, the moving speed must be lower than $0.01\text{mm}/20\mu\text{s} = 500\text{mm/s}$. Actually, in the small dynamic range near the grasping, speed is not the factor of concerned, while the accuracy is the main factor.

CONCLUSION

The investigations conducted have included the deviation detection and correction between workpiece and robot end effector by two pairs of fibre optic sensors mounted in both inner sides of gripper type robot end effector. The concept of sensor data fusion was added to enhance the robustness and self detection of the sensor system. Fuzzy linguistic variable and control rule concept were introduced to deal with the decision making and control process based on the sensor outputs. The accuracy of the integrated system shown that the method can be put into the practical application.

REFERENCE

- [1] Y.Y. Shan, W.S. Lau, C.K. Choi & Z. Zhao, Use of optical fibre sensors on robot end effector to recognize position and orientation of workpiece for flexible assembly, *Proceedings of the third Guangdong-Hong Kong Symposium on the Application of Mechatronics*, 26-27 April, 1994, Hong Kong, pp.1-5
- [2] Sony co., *High-speed Assembly Robot SRX-4CH Manuals*
- [3] Z. Zhao, W.S. Lau, A.C.K. Choi, Y.Y. Shan, Modulation functions of the reflective optical fibre sensor for specular and diffuse reflection, *Optical Engineering*, vol. 33(9), Sep., 94, pp.2986-2991

DESIGN OF A SYSTEM FOR MEASURING THE POWER DRAWN IN A STIRRED TANK WITH MULTIPLE IMPELLERS

G. Ascanio¹, B. Castro², A. Martínez², S. Chatwin² and E. Galindo²

¹Centro de Instrumentos, UNAM, Apdo. Postal 70-186, 04510 México, D.F.

²Instituto de Biotecnología, UNAM, Apdo. Postal 510-3, 62271 Cuernavaca, Mor., Mexico

ABSTRACT

The design and characterization of a system used for the power drawn in a stirred tank with multiple impellers is described. This determination is carried out by means of a strain gauges/telemetry system, which is able to measure the power drawn of each impeller. The strain gauges produce an electrical signal which is proportional to the mixing torque. This system can be operated under two sensitivity levels depending on the expecting torque values. This device was characterized in terms of the signal hysteresis, drift and precision of the measurements. The energy losses due to friction were quantified. Preliminary data of power drawn by three Rushton turbines operating in water is presented.

INTRODUCTION

The power drawn is a parameter commonly used in chemical engineering and bioprocesses. This power input is defined as the energy necessary to produce fluid movement within the container (i.e. bioreactors, mixing tanks, chemical reactors) by means of mechanical or pneumatic agitation. The accurate measurement of the power drawn to a liquid in mixing processes is necessary for scale-up procedures, process characterization, modelling mixing performance, mass transfer as well as fluid dynamic studies.

There are several methods available for the measurement of the power drawn by mixing in mechanically agitated tanks [1-3]. At industrial level, wattmeters or amperimeters are used [4-7]. At pilot plant level, dynamometers directly coupled to the motor [8], to a pneumatic [9-11], hydraulic [12] and oil lubricated bearings [13,14] are employed. Devices based on strain gauges, i.e. torque meters [15-17] and strain gauge/telemetry systems, are suitable to be used practically at all scales. The main advantage of strain gauge/telemetry systems involves the possibility of measuring the independent power drawn by each impeller of a set mounted on a shaft [18-20]. In general, commercial systems used for measuring the power drawn by mixing are designed to measure the total power input even if several impellers are mounted along the same shaft.

DESCRIPTION

The equipment was designed for experimentation at pilot level. The mixing tank was designed according to the dimensions and geometry of a 100 liter fermenter. The purpose of this equipment is to study, by analogy, the hydrodynamics characteristics into the industrial fermenters. The system consists of an acrylic tank having a water jacket for temperature control. Within the tank there is a stainless steel air sparger allowing the equipment to be operated in aerated and non aerated conditions. In order to avoid vortex formation, four baffles were attached to the tank. The tank and the motor are supported by a metallic structure. A 2 hp (1.5 kW) motor was employed and controlled through a frequency changer and then the agitation speed can be controlled between 30 and 1200

rpm. The mixing shaft is coupled to the motor by means of a coupling flange. The shaft was built in stainless steel, one inch in diameter, on which three strain gauges were mounted as well as the Rushton impellers of 1/3 of the tank diameter. In order to avoid mechanical desbalancing of the mixing shaft, it was necessary to place a bushing and a seal in the bottom of the tank. The measurement is carried out by means of a strain gauges/telemetry system, which is able to measure the power drawn of each impeller by placing strain gauges in three positions on the shaft. The strain gauges produce an electrical signal which is proportional to the mixing torque. Considering the differences in the readings of three strain gauges, the power drawn by each impeller can be measured. This system can be operated under two sensitivity levels depending on the expected torque values.

Static calibration

The static calibration gives a relationship between a defined torque and an electrical signal of the strain gauges as a voltage (V), which is obtained by means of the following equation:

$$V = M * K \dots \dots \dots (1)$$

where M is the applied torque (Nm) and K is a proportionality constant for each strain gauge, which is shown in table 1.

The torque is calculated by

$$M = F * B \dots \dots \dots (2)$$

where B is the lever arm (m), that it corresponds to ½ impeller diameter and F (kgf) is the applied force which is equal to a known weight applied tangentially to the impeller blades.

Table 1 Calibration curves for each strain gauge under two different sensitivities

Sensitivity	K(sq1)	K(sq2)	K(sq3)
Low	0.7770	0.7310	0.6573
High	3.2905	3.2743	2.8576

Measurement of energy losses due to friction

In order to avoid the shaft desbalancing, it was necessary to place a bushing and a seal in the bottom of the tank. Energy losses due to the friction with different agitation speeds (60 - 780 min⁻¹) were measured.

Since the signal of each strain gauges is little different, the torque for each strain gauge was measured and the real torque was obtained by means of the following relationship

$$M_{\text{real}} = M_{\text{measured}} - M_{\text{friction}}$$

Table 2 shows the friction values for the strain gauges. Values for strain gauges 1 and 3 are practically constant for 60 through 800 min⁻¹. The strain gauge 2 presents different torque values at different agitation speeds. The value of the torque due to the friction is the average of these readings.

Table 2. Torque due to losses friction for each strain gauge.

Strain gauge (see fig. 1)	Mfriction (Nm)
SG1	0.134
SG2	0.242
SG3	0.154

Hysteresis

The hysteresis is a measuring of the difference between two response signal, obtained in a calibration cycle. The percentage hysteresis was determined referred to the total scale, for strain gauge 2 and 3, which was 0.73% and 0.21% respectively.

Drift

The variation in the time of a response signal (drift) was determined placing weights in direction tangential to an impeller at 20.9 cm from the lower end of the shaft. The signal of strain gauge 3 was registred during 11 hours. The response was $33.6 \text{ mv} \pm 0.2 \text{ mv}$. This variation is equivalent to $\pm 0.1 \text{ Nm}$ torque, which indicates the precision limit of the instrument.

Precision

Measurements precision was determined as the percentage of variation with respect to the average of three readings for eleven different torque values. This percentage varied from $\pm 30\%$ to $\pm 2\%$ for the strain gauge 2 and $\pm 17\%$ to $\pm 1.4\%$ for the strain gauge 3. This variation was determined within a torque range of 0.33-3.77 Nm.

CONCLUSIONS

The strain gauges/telemetry equipment is a suitable system to determine the power drawn of each impeller in a stirred tank. Nevertheless, it is necessary to verify continuously the calibration, therefore its operation is delicate. The proportionality constants must be individually determined, considering that the signal of each strain gauge is little different. Taking into account this limitation, it is possible to carry out with non-Newtonian fluids, under aerated and non aerated conditions. Also, the visualization of flow patterns and the formation of air cavities is possible by means of supplementary equipment.

REFERENCES

1. Holland, I.A. and Chapman, F.S. (1966), Liquid mixing and processing in stirred tanks, Reinhold Pub., New York.
2. Nagata, S., (1975) Mixing: principles and applications, John Wiley & Sons, New York.
3. Brown, D.E., Chem. & Ind., (1977), 16, Aug. 20th, 684-688
4. Oosterhuis, N.M.G. and Kossen, N.W.F., Biotechnol. Lett., (1981), 3, 11, 645-650.
5. Nienow, A.W. et al (1994), Biotechnol. Bioeng., 44, 177-1185.
6. King, R.L. et al (1988) AIChE J., 34, 3, 506-509.
7. Asai, T. and Kono, T., J. Ferm. Technol., (1982), 60, 3, 265-268.
8. Nocentini, M., Magelli, F., Pasqualli, G. and Fajner, D., Chem. Eng. J., (1988), 37, 53-59.
9. Reséndiz, R., Martínez, A., Ascanio, G. and Galindo, E., Chem. Eng. Technol., (1991), 14, 105-108.
10. Nienow, A.W. and Lilly, M.D., Biotechnol. Bioeng., (1979), 21, 2341-2345.
11. Sánchez, A.D. et al (1992), Proc.Biochem., 27, 351-365.
12. Vránek, M., et al (1990) Int.Chem.Eng., 30, July, 562-567.
13. Machon, V., et al (1985) 5th Euro.Conf.Mixing, paper 16, 155-169.
14. Machon, V. and Vacek, J. (1985) Collec.Czech.Chem.Comm., 50, 2863-2872.
15. Abrardi, V., et al, (1990) Trans.I.Chem.Eng., part A, Nov., 516-522.
16. Böhme, G. and Stenger, M. (1988) Chem.Eng.Tecnol., 11, 199-205.

17. Brito-De la Fuente, E. et al (1991) *Chem.Eng.Res.Dev.*, **A4**, 324-331.
18. Armenante, P.M. and Li, T. (1993) *Process Mixing: Chemical and Biochemical Applications II*, *AIChE Symp. Series*, **89**, 293, 105-111.
19. Kuboi, R. and Nienow, A.W., *4th Euro. Conf. on Mixing*, (1982), **G2**, 247-261
20. Hudcova, V., Machon, V. and Nienow, A.W., *Biotechnol. Bioeng.*, (1989), **34**, 617-628.

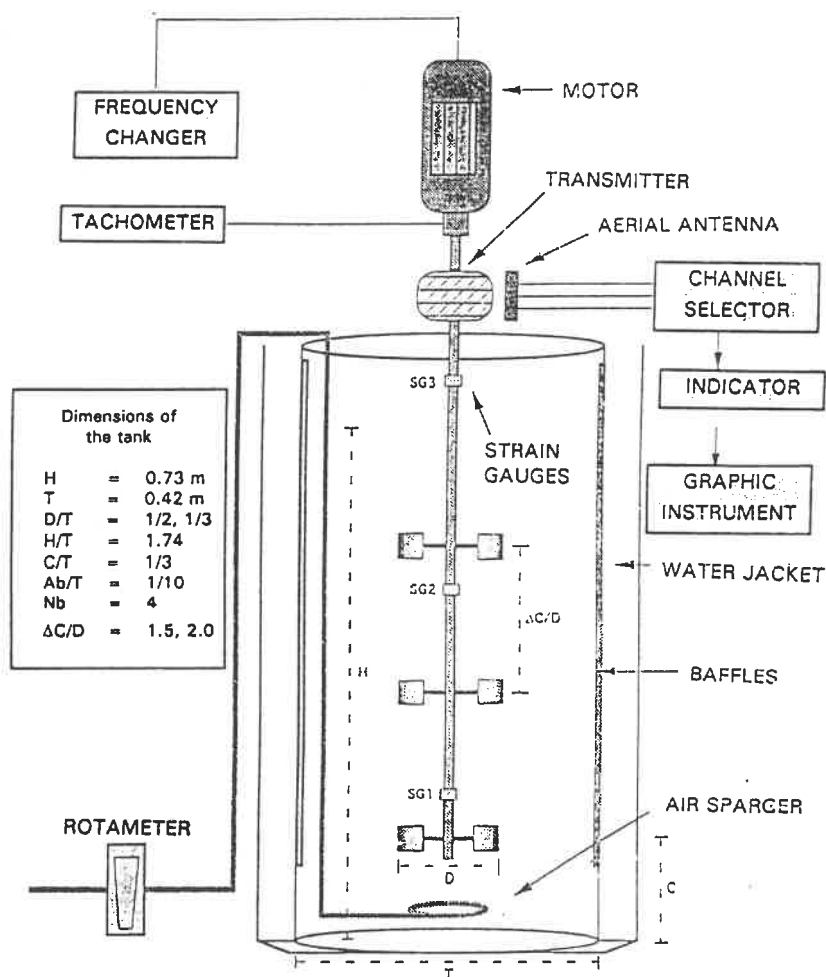


Figure 1. Diagram of the equipment for measuring the power drawn in a stirred tank with multiple impellers, (showing its parts and the employed instrumentation).

EVALUATION OF THE DIMENSIONS OF SINUSOIDAL GRATINGS

T. Ahbe, K. Hasche, K. Thiele
Physikalisch-Technische Bundesanstalt
Braunschweig und Berlin, Germany

1. Introduction

The results reported here are to be seen in the context of evaluations of measurements of the thickness of coatings and microhardness, applications of scanning force microscopy and scanning electron microscopy, the calibration of these microscopes using gratings as reference standards, and the development of methods and artefacts for this kind of calibration. This paper deals with the evaluation of the dimensions of sinusoidal gratings. The gratings to be evaluated have been manufactured as prototypes using a process consisting of three stages. In the first stage a grating was formed in photoresist using a holographic method. A copy of the photoresist grating was etched in zerodur in the second stage. In the third stage, the etched copy served as a negative for a mechanical forming process replicating gratings in PMMA. The task to be performed by us was to measure the geometry of the gratings made in the three stages and reveal deviations from a stipulated pure sinusoidal geometry, if any. The measurements were carried out with a scanning force microscope (AFM). The essential part of the evaluation was the measurement data analysis, which will be reported here.

2. The measuring instrument and its metrological characteristics

The topographical measurements of the prototyped sinusoidal gratings were carried out with an AFM suitable for metrological use [1].

Both lateral axes of the AFM were calibrated using selected sinusoidal gratings as reference artifacts. Their grating constant is about 420 nm and 480 nm. The method applied for the measurement data analysis is in principle the same as that given below [2]. For both axes we estimated an uncertainty of $u_{95} = 1 \text{ nm} + 10^{-3} \cdot l$ with $l = x$ and y , respectively. Here, the uncertainty is defined by the double amount of the corresponding standard deviation.

The data analysis also allows conclusions with respect to the orthogonality of the both axes to be drawn. The results confirmed that the non-orthogonality of the axes is sufficiently small.

The z-axis of the AFM was calibrated with step height reference standards and stair-shaped artifacts made of silicon oxide on silicon with two or three steps in the range of z up to about 800 nm [3]. The uncertainty of this calibration in the range up to 1 μm has been estimated to be $u_{95} = 1 \text{ nm} + 3 \cdot 10^{-3} \cdot z$.

3. The basic idea of the algorithm used for the measurement data analysis

With an area scan we obtain a 2-dimensional measurement data set describing the topography of the surface to be analysed:

$$d(x = \Delta x \cdot i, y = \Delta y \cdot l), \quad (i = 1, \dots, 2N; \quad l = 1, \dots, 2M)$$

The data set is approximated by a sum of non-orthogonal periodic functions describing "plane waves" reflecting the nature of the measured surface

$$f(i, l) = \sum_j f_j(i, l), \quad f_j(i, l) = k_j + c_j \cdot \sin\left(\frac{\pi}{N} \cdot n_j \cdot i + \frac{\pi}{M} \cdot m_j \cdot l + \varphi_j\right) \quad (1)$$

with the non-integer wave numbers (n_j, m_j) . The data set is also subdivided into a deterministic component $f(i, l)$ and a random residual data set $r(i, l)$:

$$d(i, l) = f(i, l) + r(i, l)$$

Instead of the common least-squares rule, a simpler approximation has been chosen, calculating the de-coupled partial functions $f_j(i, l)$:

$$\sum_{i,l} \left[d_j(i,l) - \sum_j f_j(i,l) \right]^2 = \text{Min}$$

with $d_1(i,l) = d(i,l)$ for $j = 1$ and $d_j(i,l) = d(i,l) - \sum_{p=1}^{j-1} f_p(i,l)$ for $j > 1$

According to this rule the 5 coefficients k_j , c_j , n_j , m_j and ϕ_j of a function $f_j(i,l)$ defined by (1) are calculated step by step for every value of $j = 1, 2, 3, \dots$

For every value of j , a test

$$\ln[(2MN - 1) \cdot z] < \frac{c_{nm}^2}{E[c_{nm}^2]} \quad (2)$$

must be carried out to check whether or not the data set $d_j(i,l)$ contains significant deterministic components of the type $f_j(i,l)$.

In (2) the c_{nm} signifies every Fourier coefficient belonging to $d_j(i,l)$, and $E[c_{nm}^2]$ the expectation value of the corresponding c_{nm}^2 . The value of z has a statistical significance [2].

If the criterion (2) is not fulfilled, then data set $d_j(i,l)$ is understood as a random residual data set, in which case it makes little sense to continue the approximation with further functions $f_j(i,l)$, and the procedure should be stopped:

$$r(i,l) = d_j(i,l), j = j_{\max}$$

The idea of approximation by non-orthogonal functions (1) has been described for 1-dimensional data sets in [4]. Details of this kind of analysis of 2-dimensional data sets are reported in [2].

An interesting characteristic of this algorithm is that the functions $f_j(i,l)$ are able to make the correct periodicities existing in the measured surface immediately transparent.

4. Results of a dimensional evaluation of sinusoidal gratings

The real shape of three prototypes of sinusoidal gratings should be analysed quantitatively. In the following some selected results will be reported.

A sinusoidal grating made of photoresist using a holographic method was expected to have an almost perfect shape characterised by a single 1-dimensional sinus function with a fixed value of the wavelength. A graph of the corresponding measured data set is shown in fig. 1. The calculated deterministic component of this data set is shown at the bottom left hand side of this figure. Besides the expected dominating sinusoidal wave which is characterized by the function $f_1(i,l)$ with the stipulated wavelength, an unexpected superposed higher frequency wave propagating in the orthogonal direction is visible. The random residual data set belonging to the photoresist grating is shown at the bottom right hand side of fig. 1.

The holographically-made grating served as a reference object for the etching of a corresponding grating made of zerodur. In the third stage, the etched zerodur grating was used as a negative for the mechanical pressing of a grating in PMMA material.

Figs. 2a and 2b show the deterministic components of the shape of all three gratings after low-pass filtering in the direction of m in the frequency domain, which means that the superposed higher frequency component visible in fig. 1 is now suppressed. From the coefficients of the functions $f_j(i,l)$ found for the three gratings, it may be concluded that the wavelengths corresponding to the fundamental frequencies of the gratings are 1115 nm, 1119 nm and 1119 nm, respectively. The relative uncertainty of the calculated wavelengths is estimated to be not more than 10^{-3} . The reason for the two different

wavelengths should therefore be sought in the manufacturing process. Besides the fundamental frequency of this wave, two significant harmonics were revealed for the gratings made of zerodur and PMMA. Obviously, the both harmonics result from the etching process. The AFM tip and the interaction between tip and the surface are not responsible for this effect.

The results obtained here have been used for a subsequent optimization of the manufacturing process.

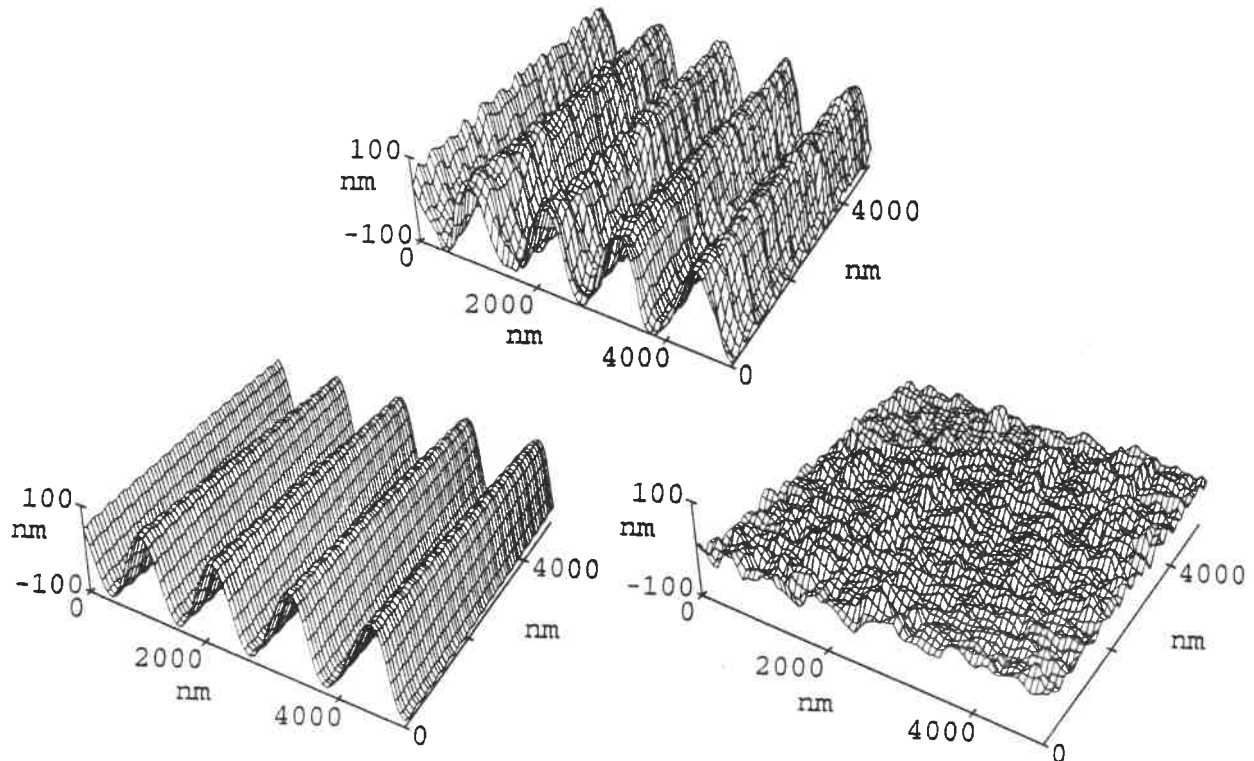


Fig. 1: A sinusoidal grating made of photoresist. Graph of the measurement data (top), its deterministic component $f(i,l)$ (at the bottom left hand side) and the remaining random data set $r(i,l)$ (at the bottom right hand side)

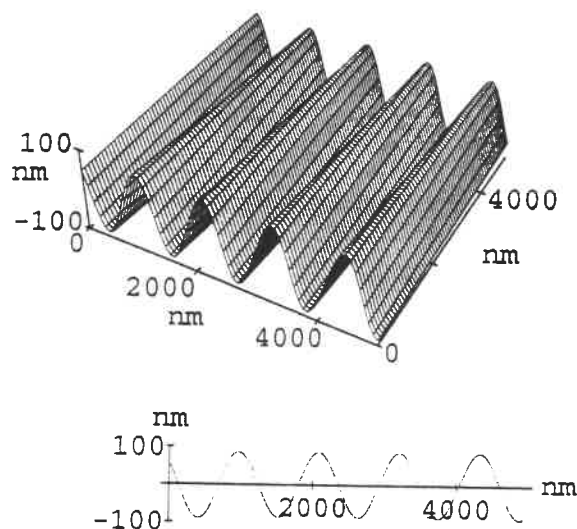


Fig. 2a: The filtered deterministic component of the grating made of photoresist and a corresponding profile

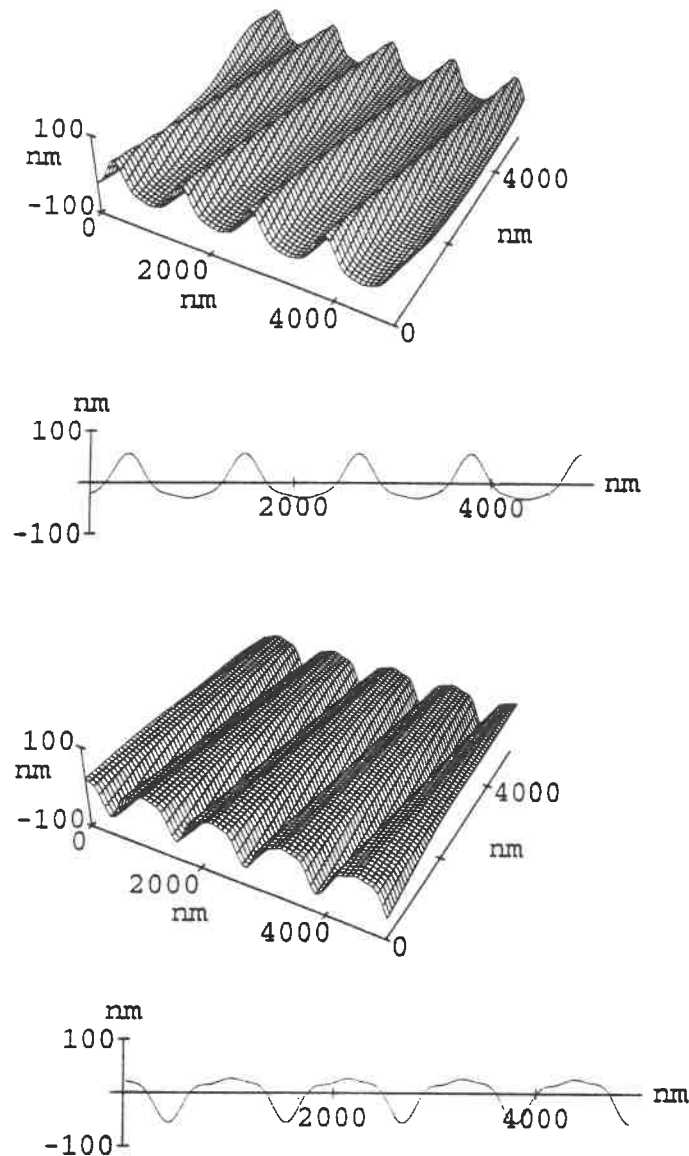


Fig. 2b: The both deterministic components of the data sets and corresponding profiles of the gratings made of zerodur (top) and PMMA resp. (bottom)

5. References

- [1] Barzke, K. et al.: Nanometer Structures Measurements with VERITEKT 3. Zeiss Information mit Jenaer Rundschau No. 4 (1994), ISSN 0941-7559, p. 10-12 (in German)
- [2] Ahbe, T.; Hasche, K.; Thiele, K.: Evaluation of Shape Measurement Data using Non-Orthogonal Periodic Functions. Proceedings of the XIII IMEKO World Congress, Torino (Italy), September 1994, p. 1635-1640
- [3] Ahbe, T. et al.: Investigations of the Thickness of Coatings and of the Geometry of Hardness Indenters by Scanning Force Microscopy. Proceedings of the 8th International Precision Engineering Seminar, Compiègne (France), May 1995, p. 207-210
- [4] Bloomfield, P.: Fourier Analysis of Time Series: an Introduction; John Wiley & Sons, New York, 1976, p. 20-25

TRACKING AND TRIANGULATION APPROACHES TO CMM PERFORMANCE VERIFICATION

P.A.C. Miguel

Centro de Tecnologia - UNIMEP; Santa Bárbara d'Oeste, Brazil; and Mechatronics Group,
School of Man. & Mech. Eng., The University of Birmingham, Edgbaston, Birmingham, UK B15 2TT.

T.G. King

Mechatronics Group, School of Manufacturing and Mechanical Engineering,
The University of Birmingham, Edgbaston, Birmingham, UK B15 2TT.

Abstract: This paper addresses the performance verification of Co-ordinate Measuring Machines (CMMs). It investigates non-contacting methods using tracking mechanisms in order to identify their potential use for checking CMMs. The paper describes different approaches using tracking systems capable of determining the position of a moveable target in space. It reviews existing techniques in this field and outlines their metrological principles. Finally, it presents a comparison between these methods to illustrate their possible application to CMM verification.

INTRODUCTION

CMM performance verification has been a relevant issue in recent years. Many different methods for checking CMMs have been suggested. A review of such methods has been provided by Miguel [1], which includes a classification into two main groups, namely artefact-based and optical techniques. The use of artefacts is relatively well documented (see for example refs. [2,3]). On the other hand, methods using optical techniques are almost restricted to the use of conventional laser interferometers. The laser interferometer does not meet all requirements for an efficient method for checking CMMs. The major requirements of an efficient method are generally processing speed, an appropriate level of accuracy, and low cost. It should also have operational flexibility and the capability of being automated. To fulfil these requirements, non-contacting techniques can be considered as an expanding field for performance verification of CMMs. Laser based techniques, for example, can be regarded as a suitable option for developing a new method.

EXISTING TRACKING SYSTEMS

Tracking systems can be categorised according to their metrological principle. They can be based on triangulation, trilateration and

spherical co-ordinate systems. Such systems usually employ laser technology using single or multiple laser beams. Single beam systems combine laser interferometric optical apparatus for measuring length and encoders for measuring angles. In essence, they are a spherical co-ordinate system and arrangements of this type have been proposed (see refs. [5-7]). Multiple beam systems consist of two or more laser beams using triangulation or trilateration. The principle of triangulation is well understood, as it has been used for centuries. The co-ordinates of a point on an object relative to some reference point can be determined if at least one distance and angles are known. Tracking systems based on triangulation usually use multiple angle information and the knowledge of the relative distance between two stations [8-10]. Laser trilateration systems which determine the co-ordinate position of a target using only distance (length) data have also been proposed [11-14].

TRACKING SYSTEMS COMPARISON

The 3-D position of a moveable target can be obtained by distance and angle measurement. Distances can be measured by laser interferometers and angles by rotary encoders. A cubic working volume with the appropriate measurement stations for three distinct principles can be seen in Figure 1.

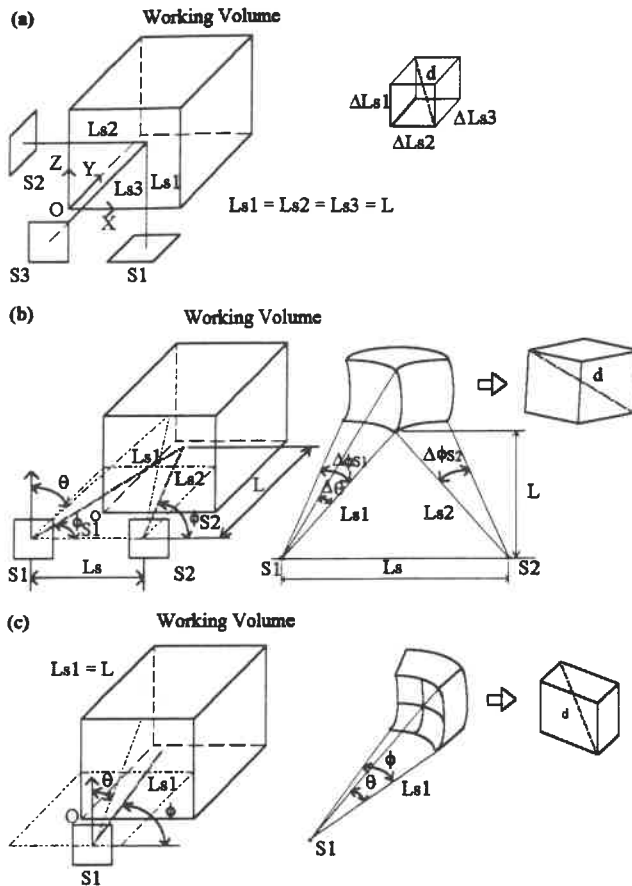


Fig. 1 - (a) Three-distance, (b) Three-angle, (c) One-distance and Two-angle System.

Figure 1(a) shows the determination of position error based only on distance measurement. In figure, S1, S2, and S3 represent three fixed measuring stations for distance measurements placed perpendicularly to planes X-Y, Y-Z, and X-Z, respectively. It is assumed the distance between the stations and the origin of the working volume is known a priori. The position error of each interferometer is ΔL_{Sji} , where i is the assessed position at the j -th measuring station. L_{S1i} , L_{S2i} , and L_{S3i} are the distances between the stations and a point at the working volume (at the centre). The volumetric error of this point can be expressed by the diagonal d , which is given by:

$$d = \Delta L_s \sqrt{3} \quad (1)$$

Figure 1(b) shows the same working volume but with two stations, S1 and S2, located in a common horizontal plane. In this case, the position of the point in the 3-D space can be

determined by triangulation, by measuring two angles in the horizontal plane and one elevation angle, assuming the distance between the two stations is known a priori. As the error is small when compared with the distance, it is possible to regard the shape of the error region as a cuboid. The diagonal d of the volumetric error at the centre of the working volume can be determined from:

$$d = \sqrt{[L^2 + (L_s/2)^2](\sin^2 \Delta \theta + \sin^2 \Delta \phi_{S1} + \sin^2 \Delta \phi_{S2})} \quad (2)$$

Finally, Figure 1(c) shows one station measuring one distance and two angles to determine the 3-D position using a spherical co-ordinate system. The distance and angle errors are given by ΔL_{S1i} , and $\Delta \theta$ and $\Delta \phi$. As the error is small when compared with the distance, it is again possible to regard the shape of the region as a cuboid. The diagonal of the cubic volumetric error can be calculated from:

$$d = \sqrt{(\Delta L_{S1})^2 + L_{S1}^2 (\tan^2 \Delta \theta + \tan^2 \Delta \phi)} \quad (3)$$

Considering the position estimation error of a He-Ne laser interferometer to be around $\pm 0.025 \mu\text{m}$ [15]) and the angular error of rotary encoders to be approximately $\pm 0.2 \text{ arc-sec}$. [16], it is possible to compare the three principles using equations 1, 2, and 3; when varying the distance L . This can be seen in Figure 2, which is valid for the centre of the working volume.

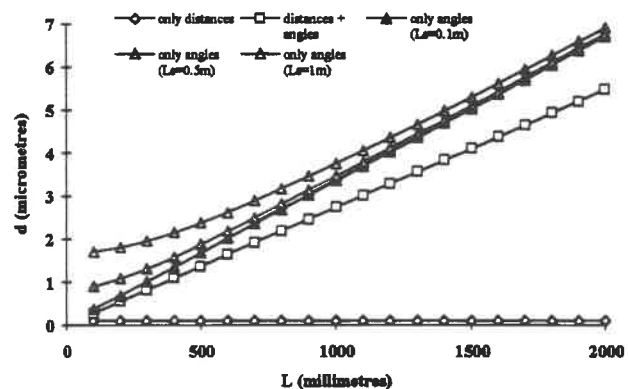


Fig. 2 - Diagonal of the Volumetric Error.

However, it would be desirable to achieve the volumetric error for each position within the working volume. To have a better understanding about the performance of these systems considering

the whole working volume, the corners of the working volume can be analysed. Then, the shape of the volumetric error at the corners of the working volume for the three-distance system will become a tetrahedron. The maximum error at the centre of working volume (calculated by equation 1), is now provided by:

$$d' = 2 \left[\sqrt{(\Delta L s_{ji})^2 + (\Delta L s_{ji} \sin \alpha)^2} \right] \quad (4)$$

Where d' is the maximum error within the tetrahedron, and it is a function of the positional accuracy ($\Delta L s_{ji}$), height (h) and angle α (Figure 3a). Although the three-distance system does not require angle data for working it is now necessary to have such information due to the angle α in equation 4. It implies that the accuracy of the system becomes indirectly dependent on the distance from the station to the target position (in this particular study). This is valid for any point within the working volume with exception of the centre position. The volumetric error shape for both three-angle and one-distance and two-angle systems have not been modified at the corners of the working volume. The diagonal d , calculated by equations 2 and 3, in this case, can still be used. The critical condition for such systems will be in the furthest position within the working volume. Figure 3b shows a working volume with 1m^3 and the results from each point (Figure 3c) using the three systems indicated in Figure 1. The stations are located 1m away from the working volume, the distance between the two stations (L_s) in the three-angle system is 1m, and the same distance and encoding resolution as cited before is used.

DISCUSSION

From these considerations, it is clear that the system based only on distance has a higher accuracy, considering the linear and angular accuracy than those based only on angles or based on distance and angles. These systems can give the position measurement regardless of the distances involved. The results from the single beam approach (one distance and two angles) were found to lie between the other two due to its partial angle dependency. The major obstacle in obtaining the

accuracy in a spherical co-ordinate system is the angular encoding errors, since the laser interferometer has appropriate displacement measuring accuracy. As expected, the system based only on angles resulted in a worse performance than the others due to its total angle dependency.

To be used for CMM verification, a measuring system should have a measuring accuracy considerably better than that of CMMs. The device's accuracy should ideally be 10 times than that of the machine (the 'Rule of Ten' [17]). According to Jiang et al. [18], alternatively, the accuracy should be at least three times higher. Existing tracking systems are often unsuitable for measuring machines, due to the higher accuracy needed when checking them.

According to this preliminary study both systems dependent on distances and hybrid systems could be used for CMM verification, using just the resolution as a criterion for volumetric error. Distance-based systems can be influenced by other sources of error, such as stability of origins, aspects of scanner mechanisms, angles of incidence at the retroreflector, beam centre and geometry, and others [10]. On the other hand, hybrid systems could be appropriate for checking CMMs if the overall accuracy were the resolution value (or close to). It is believed that might be possible to improve the accuracy of angle-based systems. This will be reported in a future paper.

Another important factor to take into consideration is the cost involved to build the system. The relatively high cost of tracking systems employing more than one station can be one of the major obstacles to their adoption. Systems which use one station (spherical system) have the advantage that only one mechanism has to be built, which can result in lower cost.

CONCLUSIONS

Laser tracking systems can be an efficient option for checking CMMs since the appropriate level of accuracy be achievable. At the present, distance-based and hybrid systems could be applied. However, the three-distance system is a more expensive solution since it needs more than one measuring station. An alternative approach

could be the application of a single-station system with better angle accuracy. Further work will involve such development in order to contribute to CMM performance verification.

REFERENCES

- [1] Miguel, P.A.C. CMM Performance Verification. MPhil thesis, Sch. of Man. & Mech. Eng., The University of Birmingham, UK, 1995.
- [2] Miguel, P.A.C. and King, T.G. Concept, Classification, and Comparison of Performance Tests in CMMs. *Int. J. of Quality & Reliability Management*, Vol. 12, No. 8, p. 50-65, 1995.
- [3] Miguel, P.A.C. and King, T.G. Standards for CMM Verification. *Measurement J.* (submitted).
- [4] Miguel, P.A.C., King, T.G. and Davis, E.J. CMM Verification - A Review of Some Existing Techniques. *Proc. of the 31st Int. MATADOR Conference*, Manchester, p. 345-350, 1995.
- [5] Loser, R. Laser Tracking System for 3D-real-time Measurements of Moving Objects. *Tech. Messen* Vol. 60, No. 5, p. 198-202, 1993.
- [6] Lau, K., Hocken, R. and Haynes, L. Automatic Laser Tracking Interferometer System for Robot Metrology. *Prec. Eng.*, Vol. 8, No. 1, p. 3-8, 1986.
- [7] Vincze, M., Prenninger, J.P. and Gander, H. A Laser Tracking System to Measure Position and Orientation of Robot End Effectors Under Motion. *Int. J. Rob. Res.*, Vol. 13, No. 4, p. 305-314, 1994.
- [8] Gilby, J.H. and Parker, G.A. Laser Tracking System to Measure Robot Arm Performance. *Sensor Review*, p. 180-184, October 1982.
- [9] Mayer, J.R. and Parker, G.A. Calibration and Assessment of a Laser Based Instrument for Robot Dynamic Calibration. 11th IMEKO Congress, Houston, p. 327-334, 1989.
- [10] Mayer, J.R. and Parker, G.A. A Portable Instrument for 3-D Dynamic Robot Measurements Using Triangulation and Laser Tracking. *Trans. Robot. & Auto.*, Vol. 10, No. 4, p.504-516, 1994.
- [11] Brown, L.B. Laser Trilateration - A New Laser Coordinate Measurement System Improves Accuracy of CMMs. *Society of Manufacturing Engineers*, Paper MS 88-481, 1988.
- [12] Laser Trilateration - A Fast, Accurate and Flexible Measuring Technique. *Quality Today*, p. 49-50, May 1989.
- [13] Greenleaf, A. H. Self-calibrating Surface Measuring Machine. *SPIE Advanced Technology Optical Telescopes*, Vol. 332, p. 327-333, 1982.
- [14] Nakamura, O. et al. T. A Laser Tracking Robot Performance Calibration System Using Ball-seated Bearing Mechanism and a Spherically Shaped Cat's-eye Retroreflector. *Rev. Sci. Instrum.*, Vol. 65, No. 4, p. 1006-1011, 1994.
- [15] Renishaw Interferometer System. Renishaw Transducer Systems Ltd, 1989.
- [16] Heidenhain. General Catalogue, March 1993.
- [17] Degarmo, E.P., Black, J.T. and Kohser, R.A. *Materials and Process in Manufacturing*. MacMillan Publishing Co., New York, 1990.
- [18] Jiang, B.C., Black, J.T. and Duraisamy, R. A Review of Recent Developments in Robot Metrology. *J. Man. Sys.*, Vol. 7, No. 4, p. 339-356, 1988.

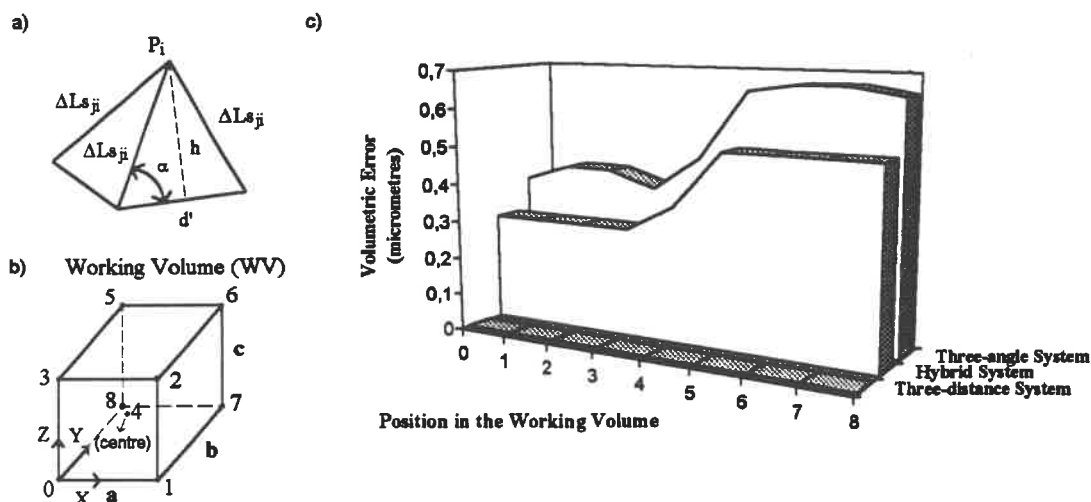


Fig. 3 - a) Volumetric Error, b) Assessed Working Volume, c) Results from the Three Systems.

A Precision Inspection System for Aircraft parts having Thin Features in CAD/CAI Integrated Environment

H.J.Pahk, W.J.Ahn

Dept. of Mechanical Design and Production Engineering
Seoul National University, Seoul, Korea

1. Introduction

The development of CAD/CAM(Computer Aided Design/Computer Aided Manufacturing) technologies have given revolutionary effects on manufacturing industry in that it has pursued optimization in the design and manufacturing of products. The CAI (Computer Aided Inspection) technique has emerged practically after introduction of computer controlled coordinate measuring machines(CMMs). There were several previous reports for the computer aided inspection technologies in pursuit of CAD/CAI integration for the manufactured parts [1,2,3,4,7]

For the parts or products, however, having very thin and sharp geometry, e.g. turbine blades, computer controlled measuring machines quite often fail to measure the thin geometry of parts, because the precision alignment for the thin dimensional parts is hardly obtained. For the tolerance evaluation, some algorithms[e.g.7] were previously proposed when the normal vector as well as position coordinates are accurately known for the measured points. However, in most cases of profile measurements, the accurate position coordinates and normal vectors of the actual measurement points are hardly obtained, thus lowering the accuracy of the analysis results.

In this paper, a new technique is proposed for the parts of very thin and sharp geometry, based on the precision alignment of the parts using a two stage alignment procedures: rough phase alignment based on the conventional 6 points probing on clear cut surfaces, and fine phase alignment using the least squares method based on the measurement feedback. For the accurate tolerance evaluation, an improved method is proposed: the actual measurement points are obtained as the closest points on the CAD geometry by the developed subdivision technique, and the Tschebyscheff norm is applied iteratively to get the optimum alignment of the measurement datum, giving the profile tolerance accurately. The developed inspection system has been applied to practical procedures of blade manufacturing in CAD/CAI environment, and demonstrated high performance. Fig.1 shows a typical CAD/CAI integration system around networks of computers and electro chemical machine for manufacturing turbine blades.

2. Reconstruction of Geometric Features for Measurement

The CAD data base of turbine blades to be inspected is accessed, and the geometric features are reconstructed. Generally, a turbine blade consists of airfoil fairing in the direction of air flow, spanwise fairing in the perpendicular direction, and a base block as the reference. The 3 dimensional representation of a turbine blade is shown in fig.2. Each airfoil section is then stacked in the spanwise direction with respect to the stacking point, where the stacking point is the reference point for generating momentum of air flow. the airfoil section is usually composed of several features: pressure surface (curve), suction surface(curve), leading edge, and trailing edge, etc. The pressure / suction curves are generally modeled as 3 dimensional smooth spline curves, and the leading and trailing edges are modeled as circles, or ellipses of very thin dimension. The quality control level of aircraft manufacturing industry are strict, and the dimensional inspection of the turbine blades is focused on the tolerance of leading and trailing edges as well as profile tolerance of pressure/suction surfaces.

For practical turbine blades, chord length based 3 dimensional spline curves are generally used for modeling the pressure/suction surfaces.

3. Rough Phase Alignment Procedure for Initial Measurement

In order to perform the CMM measurement based on CAD defined geometry, the relationships between the CAD geometry and the CMM coordinate system have to be found, and the traditional 6 points method has been used for the coordinate alignment. Let D as the workpiece coordinate data, M as the CMM coordinate data. The relationship between the coordinate data can be expressed as the transformation matrix, T_1 , such that $M=T_1 D$ is satisfied.

$$T_1 = \begin{bmatrix} t_{11} & t_{12} & t_{13} & t_x \\ t_{21} & t_{22} & t_{23} & t_y \\ t_{31} & t_{32} & t_{33} & t_z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (4)$$

where t_{ij} ($i,j=1,2,3$) are responsible for rotation; t_x, t_y, t_z are for the translational offset between the coordinate data. The component of T_1 matrix can be obtained by several methods, and the conventional 6 points probing has been used: 3 points, 2 points, 1 point probing on the clear cut surfaces of the reference block of blade. The CNC(Computer Numerical Control) code for CMM measurement can be obtained by multiplication of the T_1 matrix and the CAD defined workpiece coordinate data.

$$M = T_1 (D+rN) \quad (5)$$

where D is the workpiece coordinate, r is the probe radius of the CMM, and N is the unit normal vector at the target points, which is to consider the outward offset points for the measurement target points. Therefore rough phase alignment is performed, and the CNC code is generated in widely accepted machine code format such as DMIS (Dimensional Measuring Interface Specification); then initial measurement procedure is performed on the curved parts of blade, after the CNC code is being downloaded into the CMM.

4. Fine Phase Alignment Based on Iterative Least Squares Technique

Although the alignment is performed based on the 6 points probing on the reference block, the CMM often fails to measure the parts of thin geometry, and this is because that there may exist misalignment error due to the form error of the reference block and the error at the time of probing. The misalignment error can be evaluated and compensated by introducing a second transformation matrix, T_2 . When the measured data, MM , of the curved part are converted to the nominal CAD coordinate system, they can be compared with those of nominal CAD data, and T_2 links the relationships between the nominal CAD data and the converted measurement data of the parts. The measured data can be converted to the nominal CAD coordinate system, by multiplying T_1^{-1} to the measured data, MM .

$$T_1^{-1}MM = T_2(D+rN) \quad (6)$$

The T_2 matrix of 4X4 can be determined by the least squares technique, which minimizes the sum of squares of distance between the converted measurement data ($T_1^{-1}MM$) and the nominal CAD data($D+rN$),

$$E = \sum |T_1^{-1}MM - T_2(D+rN)|^2 \quad \text{for all measurement points} \quad (7)$$

The variational principle can be applied to solve the components of T_2 matrix, and one of convenient

method for the evaluation of T_2 matrix is to use the pseudo inverse technique for computational efficiency. Therefore T_2 matrix can be evaluated as follows

$$T_2 = T_1^{-1}MM(D+rN)^T[(D+rN)(D+rN)^T]^{-1} \quad (8)$$

The calculated T_2 matrix is now used to generate the updated CNC codes, downloaded into the CMM, and the second measurement data, $MM^{(2)}$ are obtained. When the generated measurement path still fails to measure the thin section of parts, a further iterative procedure can be introduced. Based on the second measurement data $MM^{(2)}$, a second $T_2^{(2)}$ matrix can be evaluated. Prior to going to further iterations, ERR, the sum of squares of distance can be used as a criterion. That is,

$$\begin{aligned} \text{ERR} &\leq \text{TOL} \\ \text{where ERR} &= \text{sum of squares of distance} \\ &= \sum |T_1^{-1}MM^{(i)} - T_2^{(i-1)}(D+rN)|^2 \end{aligned} \quad (9)$$

and $MM^{(i)}$ is the i th measurement data, and $T_2^{(i-1)}$ is the $(i-1)$ th generated T_2 matrix.

When the criterion is not met, further iterations can be performed, and the finer alignment is possible. Fig.3 shows the flowchart for the developed system using rough and fine phase alignment. Fig.4 shows the distance deviation between the nominal CAD and actual points of the measured parts, where the distance deviation is significantly reduced at the fine alignment stage, and it is found that the fine phase alignment is performed after the first iteration. Therefore a precision alignment technique is proposed, and demonstrated its efficiency in practical manufacturing of turbine blades having very thin sections.

5. Tolerance Evaluation

Tolerance Evaluation Algorithm

The measured data are to be analysed in terms of tolerance specification of the blade parts: profile tolerance for pressure and suction sides, the radial tolerance of the leading and trailing edges. When curved parts are measured with CMMs, the actual measurement points and actual normal vectors are in general slightly different from the nominal points and nominal normal vectors. The reasons are possibly the form errors in parts and machines, and deviation in approaching directions, etc. Thus the actual measurement points have to be determined on the CAD coordinate system, and the closest points on the CAD coordinate system corresponding to the measured points can be good approximation. An efficient subdivision algorithm for finding the closest points has been developed in this paper using an

iterative method.

In order to evaluate the profile tolerance, the setup position of the reference profile is to be determined such that the Tschebyscheff norm be minimum. In this paper, an iterative algorithm has been implemented using the evaluation norm between the measured points and the corresponding closest points. In order to calculate the profile tolerance in terms of the Tschebyscheff norm, the optimum setup of the measured points datum with respect to the CAD datum is to be determined such that the Tschebyscheff norm be minimum. When the optimum setup, T_o , can be determined, the profile tolerance can be calculated as the maximum deviation between the measured points and the CAD datum.

Let M_i be the measured points data, and P_i be the calculated closest points on the CAD datum. When a transformation matrix, T , between the measured points data and the closest points data is known, the Tschebyscheff norm of power p , L_p , can be defined.

$$L_p = [\sum |M_i - TP_i|^p]^{1/p} \quad (10)$$

The L_p can be minimised with respect to the transformation matrix, T , and the optimum transformation matrix, T_o , can be found. The profile tolerance, E , is then

$$E = \max |M_i - T_o P_i| \text{ for } i=1,2..N \quad (11)$$

In this paper, an algorithm for the profile tolerance evaluation is implemented, and the flowchart is shown in fig.5.

Tolerance Evaluation Results

The developed tolerance evaluation module is applied to practical turbine blade manufacturing processes. Fig.6 show the maximum deviation (profile tolerance) as the iteration number increases. When the power of T norm increases in the fig.7, the rate of convergence slows down, and better values of profile tolerance are obtained at the cost of longer computing time. For a typical section of blade, the profile tolerance values are 0.168mm, 0.171mm for the pressure side and suction side, respectively, while the nominal profile tolerance was 0.200mm for both sides. For the radius of the leading and trailing edges, the mean radius is calculated for both edges based on the measured data points. The radii have been calculated as 0.148mm, 0.154mm for the leading and trailing edges, respectively, while the nominal radii are 0.150 +/- 0.05mm for both.

6. Conclusion

The developed method provides useful techniques for the on-line quality control in the manufacturing

process of turbine blades, and has demonstrated its performance in practical application. The conclusion is:

- (1) A precision alignment technique is developed, which is rough phase alignment by 6 points probing, and then followed by fine alignment using iterative least squares technique, based on the measurement feed back. The developed alignment technique is found useful especially for the measurement of parts having very thin sections, like aircraft parts.
- (2) CAD/CAI integration technique is found essential for measurement and inspection of complicated parts having thin sections, and the CAD defined curved or complicated geometry can be used for the reference for alignment.
- (3) An efficient and iterative algorithm has been developed for the profile tolerance evaluation, based on the Tschebyscheff norm between the measurement points data and the closest points data on the CAD defined geometry.
- (4) A CAD/CAI integrated precision inspection system has been practically implemented around computers-manufacturing facilities : data interface from CAD station, initial and fine measurement path plan in DMIS format, measurement operation, tolerance evaluation based on the Tschebyscheff norm, and the results documentation. The developed method provides useful techniques for the on-line quality control in the manufacturing process of turbine blades, and has demonstrated its performance in practical application.

References

1. Duffie,N., Bollinger,J.,Piper,R. and Kroneberg, M. CAD directed Inspection and Error Analysis Using Surface Patch Databases, Annals of CIRP,Vol 33,1984
2. Bojanic,O.P.,Majstrovic,D.V., and Milacic,R.V., CAD/CAI Integration with Special Focus on Complex Surfaces, Annals of CIRP,Vol41,1992
3. Menq,C.H.,Yau,H.T.,andLai,G.W., Automated Precision Measurement of Surface Profile in CAD Directed Inspection,IEEE Transactions on Robotics and Automation,Vol 8,No.2,1992
- 4.Pahk,H.J., Kim,Y.H., Hong,Y.S., and Kim,S.G., Development of Computer Aided Inspection System with CMM for Integrated Mold Manufacturing, Annals of CIRP, Vol42/1/1993
- 5.Choi,B.K., Surface Modeling for CAD/CAM, Elsevier, 1991
6. Dimensional Measuring Interface Specification Ver2.1, Computer Aided Manufacturing - International, Inc., 1989.
- 7.Goch,G., and Tschudi,U., A Universal Algorithm for the Alignment of Sculptured Surfaces, 597-600, Annals of CIRP 41/1/1992.

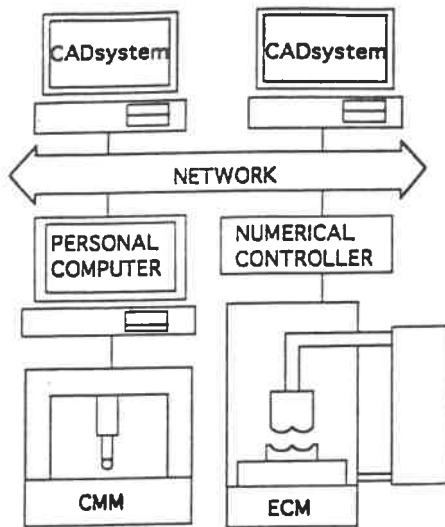


Fig.1 CAD/CAI Integration System around Networks of Computers and Machines

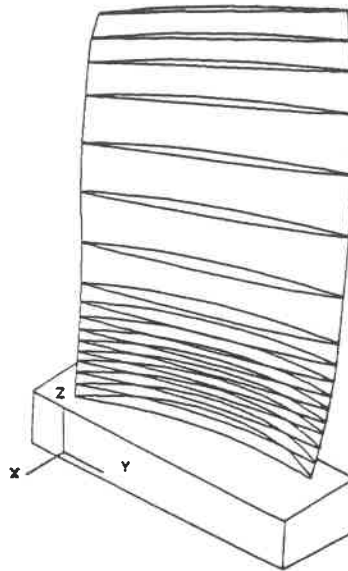


Fig.2 Three Dimensional Representation of Turbine Blade

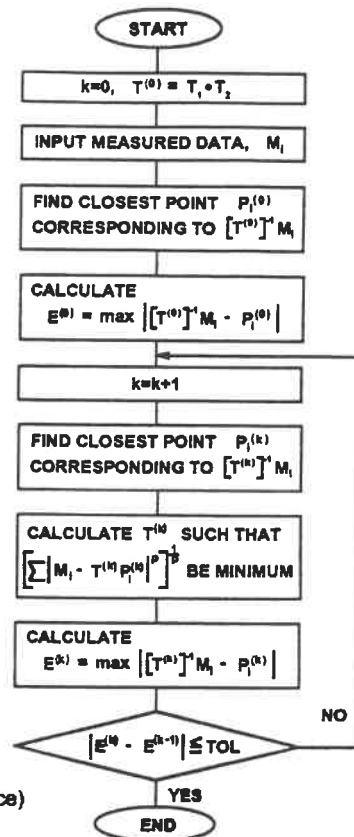


Fig.5. Flowchart of the Profile Tolerance Algorithm Based on the Iterative Tschebyscheff Norm

Fig.6 Maximum Deviation (Profile Tolerance) vs Iteration Number

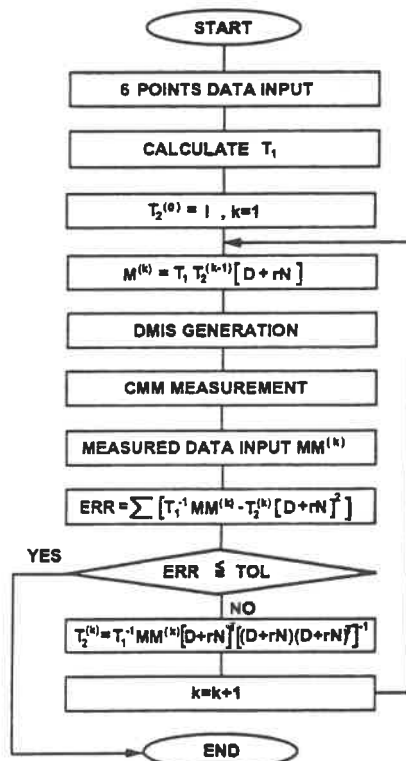
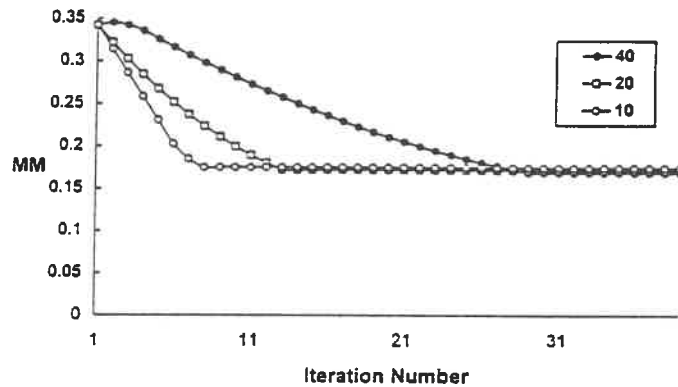
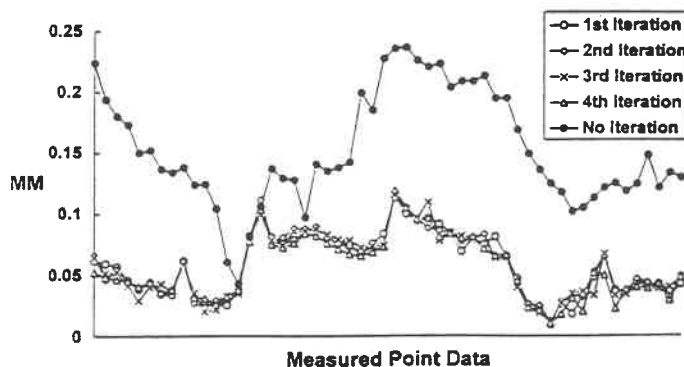


Fig.3 Flowchart for the Developed System Using Rough and Fine Phase Alignment

Fig.4 Distance Deviation between the Nominal and Actual Points



A New Method of Evaluating the Form Error of Torus

Shugui LIU Yugang LI Zhihua ZOU & Jing LI
Tianjin University, P. R. China

Torus parts are widely used in industrial fields. To measure them and evaluate their form errors, the authors put forward a new method, based on the least squares principle in this paper. Through detailed study by theoretical analysis and extensive computer simulation, it is proven that this method is accurate and quick. The method has a great value of practical usage also, for it can be easily adapted to the widely used coordinate measuring machine (CMM) and other special measuring machines.

Keywords: torus, least squares method, linear regression, form error evaluation.

INTRODUCTION

The torus is one of the basic geometric elements, and it is an important exponent of the revolution surface. Today, tori are widely used in industrial fields, especially in rotating parts. The inner and the outer raceway grooves of a ball bearing, for example, are torus shape (this shape can also be easily seen in many other places). Therefore, it is very important to develop an effective method to evaluate tori.

But to date, no satisfactory method for the evaluation of torus form errors has been developed. Traditionally, a torus is measured as a series of circles, and its form error is estimated according to the evaluated results of all the circles, but the crux is, torus is a whole element, and it should be measured and evaluated as a whole part, not as several separated ones.

To evaluate tori properly, the authors provide a new method in this paper that evaluates a torus as a whole part.

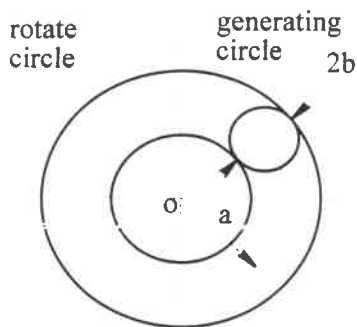


Fig.1 Outlook of torus

REGRESSION ANALYSIS

1. Analytic function of torus.

When a generating circle whose radius is b rotates about an axis $\vec{D}(l\vec{i} + m\vec{j} + n\vec{k})$, a torus is generated. In machine coordinate system S_0 , a randomly placed torus has the following analytic function:

$$\left[(x - x_0)^2 + (y - y_0)^2 + (z - z_0)^2 + a^2 - b^2 \right] = 4a^2 \left\{ (x - x_0)^2 + (y - y_0)^2 + (z - z_0)^2 - [l(x - x_0) + m(y - y_0) + n(z - z_0)]^2 \right\} \dots\dots\dots (1)$$

In order to evaluate this 7-dimensional element, least squares method is applied here.

2. least squares regression

When a group of points $p_i(x_i, y_i, z_i) (i = 1 \dots N, N \geq 7)$ is measured on the surface of a torus part, there is a deviation e_i between each point p_i and the surface of a best fitted torus.

$$E_s(a, b, l, m, x_0, y_0, z_0) = \sum_{i=1}^n e_i^2 \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \quad (2)$$
$$e_i(a, b, l, m, x_0, y_0, z_0) = \left(-2a \left\{ (x_i - x_0)^2 + (y_i - y_0)^2 + (z_i - z_0)^2 - [l(x_i - x_0) + m(y_i - y_0) + n(z_i - z_0)]^2 \right\}^{1/2} + (x_i - x_0)^2 + (y_i - y_0)^2 + (z_i - z_0)^2 + a^2 \right)^{1/2} - b \dots \dots \dots (3)$$
$$\begin{aligned} \partial E_s / \partial x_0 &= 0, \partial E_s / \partial a = 0, \partial E_s / \partial l = 0, \\ \partial E_s / \partial y_0 &= 0, \partial E_s / \partial b = 0, \partial E_s / \partial x_0 = 0, \dots \dots \dots (4) \\ \partial E_s / \partial z_0 &= 0 \end{aligned}$$

Through initial calculating, the center and the direction of the torus in the machine coordinate system S_0 can be easily estimated. Workpiece coordinate system S_1 is established according to the estimated value: The origin of S_1 is set to the estimated torus center $\hat{O}_1$, and its z-axis is set to be parallel to the estimated torus direction $\hat{D}_1$. The x-axis (l_x, m_x, n_x) and the y-axis (l_y, m_y, n_y) of S_1 are determined by the z-axis and the additional condition $n_x = 0$.

step 1. Solve the linerized equation and calculate each parameter on the current workpiece coordinate system S_i .

step 2. Establish workpiece coordinate system S_{j+1} on S_j as how S_1 is established on S_0 .

step 3. Convert the system coordinate of each measured point $p_i(x_i, y_i, z_i)$ to S_{i+1} and redo step 1.

89

iterative computation is introduced: When the calculated torus center $O_j(x_j, y_j, z_j)$ in coordinate system S_j and the torus direction $D_j(x_j, y_j, z_j)$ satisfies the conditions of $|x_j| < \varepsilon_1, |y_j| < \varepsilon_1, |z_j| < \varepsilon_1, |l_j| < \varepsilon_2$, and $|m_j| < \varepsilon_2$, the satisfactory accuracy is gained.

The linearized equation mentioned above is as follows:

$$\begin{bmatrix} \sum_{i=1}^N a_{i0}^2 & \sum_{i=1}^N a_{i1} \cdot a_{i0} & \cdots & \sum_{i=1}^N a_{i6} \cdot a_{i0} \\ \sum_{i=1}^N a_{i0} a_{i1} & \sum_{i=1}^N a_{i1}^2 & \cdots & \sum_{i=1}^N a_{i6} \cdot a_{i1} \\ \cdots & \cdots & \cdots & \cdots \\ \sum_{i=1}^N a_{i6} \cdot a_{i0} & \sum_{i=1}^N a_{i6} \cdot a_{i1} & \cdots & \sum_{i=1}^N a_{i6}^2 \end{bmatrix}_{7 \times 7} \times \begin{bmatrix} w_0 \\ w_1 \\ \vdots \\ w_6 \end{bmatrix}_{7 \times 1} = \begin{bmatrix} \sum_{i=1}^N a_{i0} b_i \\ \sum_{i=1}^N a_{i1} b_i \\ \cdots \\ \sum_{i=1}^N a_{i6} b_i \end{bmatrix}_{7 \times 1} \quad \cdots \cdots \cdots (5)$$

Where: $w_0 = x_0$, $w_1 = y_0$, $w_2 = z_0$, $w_3 = a^2 - b^2$, $w_4 = a$, $w_5 = l \cdot a \cdot n$, $w_6 = m \cdot n \cdot a$,
 $a_{i0} = 2ax_i / (x_i^2 + y_i^2)^{1/2} - 2x_i$, $a_{i1} = 2ay_i / (x_i^2 + y_i^2)^{1/2} - 2y_i$, $a_{i2} = -2z_i$, $a_{i3} = 1$,
 $a_{i4} = -2(x_i^2 + y_i^2)^{1/2}$, $a_{i5} = 2x_i z_i / (x_i^2 + y_i^2)^{1/2}$, $a_{i6} = 2y_i z_i / (x_i^2 + y_i^2)^{1/2}$,
and $b_i = -x_i^2 - y_i^2 - z_i^2$

3. Error evaluation

When all points of a toroidal surface lie between two concentric tori of the rotating diameter $2a$, the smallest difference of the small diameter b is the torus form error. Then the form error can be evaluated as : $\delta = e_{\max} - e_{\min}$.

SIMULATION RESULTS

With computer aid, the data and their errors of a measured imaginary torus can be generated.

By evaluating these tori, the accuracy and efficiency of this evaluation method is tested. The method of data error generation is illustrated in Fig.2; That is, the radius of generating circle b is added with

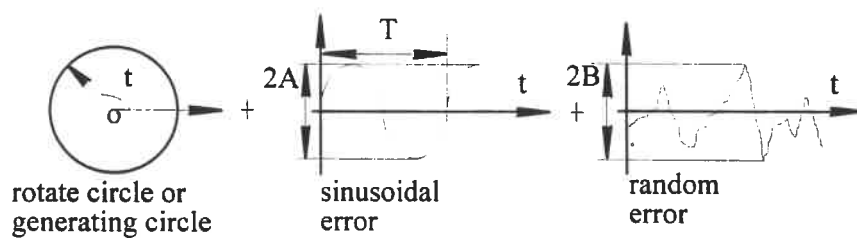


Fig. 2 Error adding method

sinusoidal error e_{c1} and hypodispersion error e_{r1} and the radius of rotate circle a is added with sinusoidal error e_{c2} and hypodispersion error e_{r2} . Thus the system coordinate of each measured point p_i is function of $(a + e_{c2} + e_{r2}, b + e_{c1} + e_{r1})$. From numerous of simulation experiments, we illustrate the test result of this method with the imaginary torus that is added to the rotate circle and generating circle with sinusoidal error whose

$T = \pi/2$, $A = 4\mu m$, and with hypodispersion error whose $B = 1\mu m$ as an example. Others almost have the same test result.

Fig.3 shows the relationship between the deviation E_s and the iteration times j of the imaginary torus.

It is clearly shown in Fig.3 that by increasing the number of iterations, the solution gets gradually better, but it becomes less effective when the iteration times are more than 6 or 7. So in simulation experiments, the convergence criterion is always set $\varepsilon_1 < 10^{-9}$ meter and $\varepsilon_2 < 10^{-12}$, since the resolution of the transducer is lower than 10^{-7} meter. As an example of the calculation accuracy of this method,

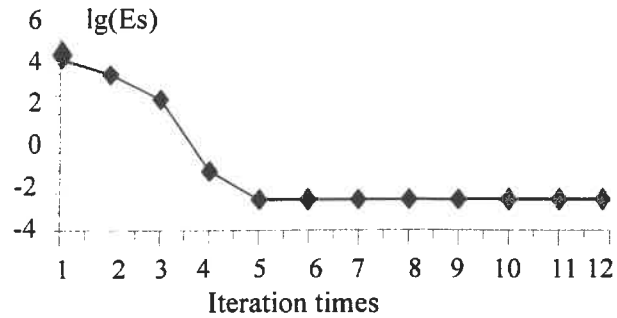


Fig.3 iteration efficiency

Fig.4 shows the $(e_{ai} - e_{bi})$ for 480 measured points. Here, e_{ai} is the given error of each point, which is the combination of $4\mu m$ sinusoidal errors e_{c1}, e_{c2} and $1\mu m$ hypodispersion error e_{r1}, e_{r2} when generating torus. The e_{bi} is the error of each point calculated

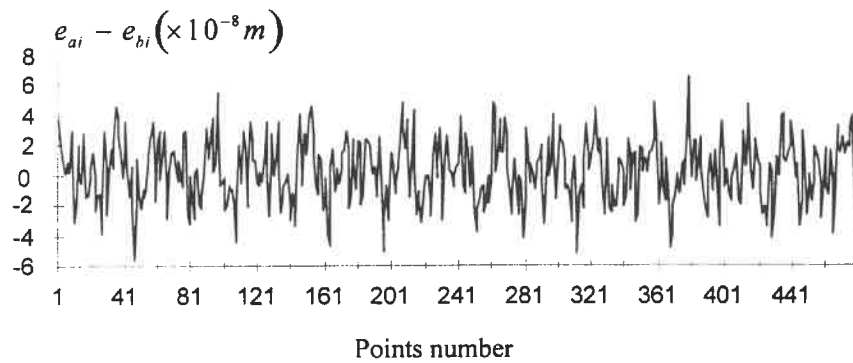


Fig.4 Accuracy of this method

using this method. It is clearly shown in Fig.4 that each e_{ai} and e_{bi} is almost equal, and the standard deviation of $(e_{ai} - e_{bi})$ is 2.53×10^{-8} meter; The average value of $(e_{ai} - e_{bi})$ of the 480 points is 2.95×10^{-9} meter.

CONCLUSIONS

The main conclusions are as follows:

- The linearized least-squares-method algorithm used in this method is an accurate and quick method. Using it, the convergence criterion $\varepsilon_1 < 10^{-9}$ meter and $\varepsilon_2 < 10^{-12}$ meter can be easily achieved within six or seven iteration times. When the number of measured points is less than 100, it only takes less than 0.12s to fit a torus with the accuracy mentioned above on a 386 personal computer with a 387 numeric processor.
- This method is very prosperous in practical usage, because the measured data are recorded as the Cartesian coordinates in three dimensions, the methods can be successfully applied to the widely used three-dimensional coordinate measuring machines, and it is also useful in the evaluation of the position errors of torus surface parts. For example, it can be used to evaluate the position errors of the inter ring of a ball bearing.

Evaluation of the Position Error of Two-cylinder-workpiece Applying Maximum Material Principle

Shugui LIU , Jing LI, Zhihua ZOU & Yugang LI

Dept. of Precision Instrument Engineering, Tianjin University,
Tianjin 300072, P.R.CHINA

A new method for the evaluation of the position error of two cylinders' axes applying the maximum material principle (MMP) is presented in this article. The mathematical model considering the two cylinders as a whole and using least-squares fitting algorithm is established to evaluate various sizes of workpiece by the personal computer precisely and quantitatively. The data are acquired by means of coordinate measuring machine (CMM). Many advantages can be obtained by using this method instead of the comprehensive gage method. Its application in practical measurement can replace the material gage with a virtual one. The spread of the method will be helpful in making the position error evaluation perfect.

Keywords: maximum material principle , position error evaluation, least-squares fitting

List of symbols

$(x_1, y_1, 0)$	Coordinate of a point on the axis of virtual cylinder No.1
$(x_2, y_2, 0)$	Coordinate of a point on the axis of virtual cylinder No.2
(l, m, n)	Direction cosine of the axis of the virtual cylinders
D_{1m}	Actual maximum (for cylinders)/ minimum (for holes) radius of cylinder (or hole) No. 1 being measured
D_{2m}	Actual maximum (for cylinders)/ minimum (for holes) radius of cylinder (or hole) No.2 being measured
D_1	Maximum material size of cylinder No.1
D_2	Maximum material size of cylinder No.2
dc_0	Nominal centre distance
dc	Actual centre distance
T_0	Position error tolerance

Introduction

In modern industrial production, the wide spread of the flexible manufacturing system makes it necessary to find out a certain method so as to test the position error of the workpiece automatically and quantitatively. Unfortunately, there is no ideal method in the practical evaluation of the position error of two cylinders' axes based on the maximum material principle established till now, although such position errors are already stipulated in both International Standard (ISO1101/II-1974) and the National Tolerance Standard (GB1183-80). The authors focused on finding a good method to solve this problem.

In this method, the two cylinders are considered as a whole, on this basis, the least-squares fitting algorithm is used to establish the relevant mathematical model.

This method is much more flexible than the comprehensive gage method which one gage can only measure the cylinders of one size. The Broyden method for the solution of the non-linear equations is applied

Establishment of mathematical model

In accordance with the MMP, the axes of the two ideal cylinders to be fitted are

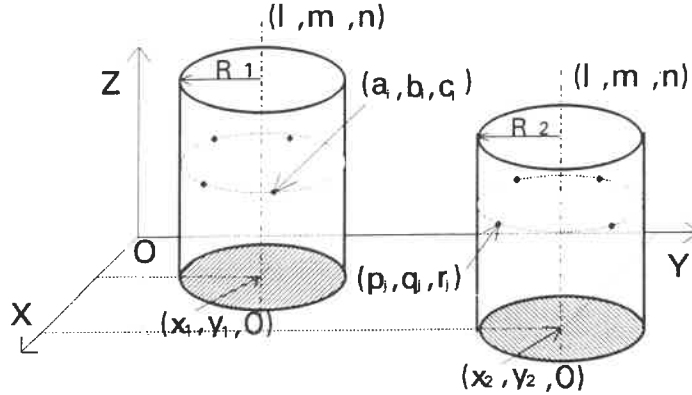


Fig.1: The mathematical model

assumed to be parallel. While neither of them is parallel to the X-Y plane, there must be a point with coordinate $z = 0$ on the axes. The direction (l, m) and locations (x_1, y_1, x_2, y_2) of the axes and the radii (R_1, R_2) of the two cylinders are considered as the parameters of these two cylinders. On the basis of these parameters and the points measured on each cylinder, the distance d_{1i} between the i th measured point (a_i, b_i, c_i) on

cylinder No.1 and its axis can be expressed as :

$$d_{1i} = \{[(a_i - x_1)^2 + (b_i - y_1)^2 + c_i^2 - \{l(a_i - x_1) + m(b_i - y_1) + c_i(1 - l^2 - m^2)^{1/2}\}^2]\}^{1/2} \quad (1)$$

The deviation e_{1i} between d_{1i} and the ideal radius R_1 of cylinder No.1 is :

$$e_{1i} = d_{1i} - R_1 \quad (2)$$

The distance d_{2j} between the j th measured point (p_j, q_j, r_j) on cylinder No.2 and its axis is:

$$d_{2j} = \{[(p_j - x_2)^2 + (q_j - y_2)^2 + r_j^2 - \{l(p_j - x_2) + m(q_j - y_2) + r_j(1 - l^2 - m^2)^{1/2}\}^2]\}^{1/2} \quad (3)$$

The deviation e_{2j} between d_{2j} and the ideal radius R_2 of cylinder No.2 is :

$$e_{2j} = d_{2j} - R_2 \quad (4)$$

Now that the deviations of the measured points to the relevant ideal cylinder are available, considering the two cylinders as a whole, the sum of the squares of the total

$$\text{deviation is: } F(l, m, x_1, y_1, x_2, y_2, R_1, R_2) = \sum_{i=1}^{k_1} e_{1i}^2 + \sum_{j=1}^{k_2} e_{2j}^2 \quad (5)$$

Where, k_1 and k_2 are the number of points measured on each cylinder respectively.

From the theorem of the extreme value, when $F(l, m, x_1, y_1, x_2, y_2, R_1, R_2)$ reaches the minimum value, differentiating F partially with respect to $l, m, x_1, y_1, x_2, y_2, R_1$ and R_2 are zero.

$$\begin{aligned} \frac{\partial F}{\partial l} = 0; \quad \frac{\partial F}{\partial m} = 0; \quad \frac{\partial F}{\partial x_1} = 0; \quad \frac{\partial F}{\partial y_1} = 0; \\ \frac{\partial F}{\partial x_2} = 0; \quad \frac{\partial F}{\partial y_2} = 0; \quad \frac{\partial F}{\partial R_1} = 0; \quad \frac{\partial F}{\partial R_2} = 0 \end{aligned} \quad (6)$$

The solution of the non-linear equation set gives the value of the parameters of the virtual gage. This equation set is solved by Broyden method.

Evaluation process based on MMP

After these equations have been solved, D_{1m}/D_{2m} can be calculated by taking the maximum or minimum value of d_{1i}/d_{2j} , and $dc_0 = [(x_2 - x_1)^2 + (y_2 - y_1)^2]^{1/2}$. On condition that the dimension of the workpiece is within the tolerance, according to the MMP:

For cylinders, if $(D_{1m} + D_{2m}) \leq (D_1 + D_2)$, the compensation value ΔT of the position tolerance gained from the size tolerance is:

$$\Delta T = (D_1 - D_{1m}) / 2 + (D_2 - D_{2m}) / 2 = [(D_1 + D_2) - (D_{1m} + D_{2m})] / 2 \quad (7)$$

For holes, if $D_{1m} + D_{2m} \geq D_1 + D_2$, the compensation value ΔT is:

$$\Delta T = (D_{1m} - D_1) / 2 + (D_{2m} - D_2) / 2 = [(D_{1m} + D_{2m}) - (D_1 + D_2)] / 2 \quad (8)$$

$$\text{then the new permitted position error } T_1 = \Delta T + T_0. \quad (9)$$

Either for cylinders, if $(D_{1m} + D_{2m}) > (D_1 + D_2)$, or for holes, if $(D_{1m} + D_{2m}) < (D_1 + D_2)$, the workpiece exceeds the specification limit of size tolerance.

If $|dc - dc_0| < T_1$, the workpiece is acceptable. Otherwise, it is defective.

Computer simulation

The computer simulation is designed to verify the effectiveness of the algorithm. The simulated data are generated based on the mathematical model, the initial value of every parameter can be given artificially or through certain algorithm. The parameters of the virtual gage, computed by processing the data are compared with these of the

generated cylinders to test the precision of our algorithm. In the experiment, the precision can reach 10^{-6} mm only through 10 times of iteration.

A Broyden method is used to solve the non-linear equations. The solution that meets the precision requirement can be obtained in no more than 0.2 second by a PC 386DX/40 if there are 100 points on each cylinder. The experimental results show that the method is accurate and efficient. The Euclidean

norm $(S = (\sum_{i=0}^7 f_i^2)^{1/2})$, $f_0, f_1, \dots, f_7$ are

the partial differentiations of formula (5)), the off centre of the two

cylinders ($e_1 = [(x_1^{(k)} - x_1)^2 + (y_1^{(k)} - y_1)^2]^{1/2}$, $e_2 = [(x_2^{(k)} - x_2)^2 + (y_2^{(k)} - y_2)^2]^{1/2}$) and the tilting angle A between $(l^{(k)}, m^{(k)}, n^{(k)})$ and (l, m, n) converged quickly in the process of iteration, as shown in fig.2, where to facilitate figuring, the logarithm is taken. The computer programs were written in C language.

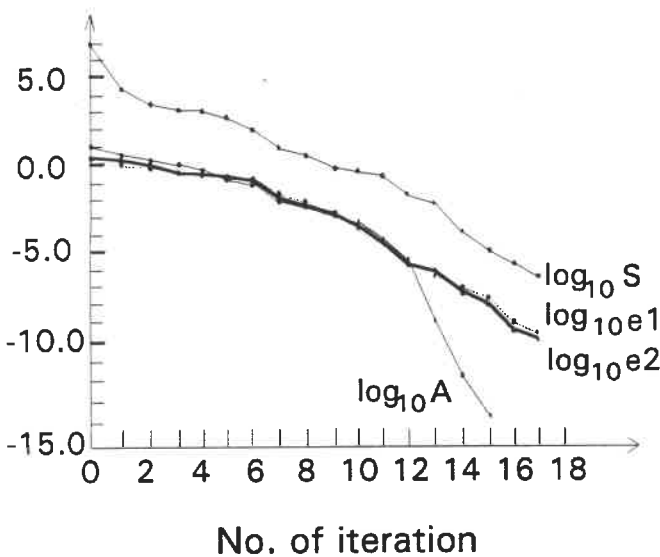


fig.2 convergence speed

Experimental results

The CMM (TOKYO XYZA 16S) is chosen as the measuring instrument in the experiment. The rod bar is used as the workpiece (see figure 3), the steps of measuring

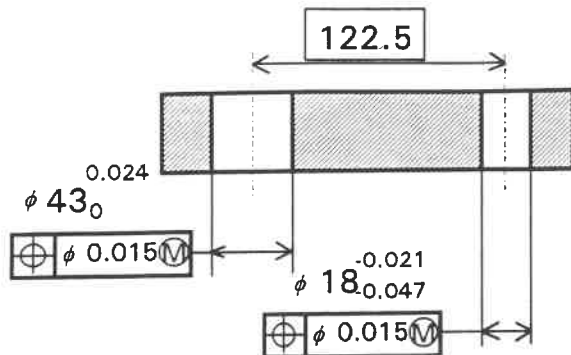


Fig 3: rod bar

process are as follows:

- Establish the coordinate system of the workpiece.
- Calibrate the radius of the probe to avoid the error caused by it.
- Measure the points on the two holes on the rod bar.
- Process the data to determine the value of the parameters using the mathematical algorithm.

Rotate and turnover the workpiece to make it measured at 7 locations separately, then recycle the above steps. The standard

deviations of multiple measurement values are small enough as shown in Table 1, so the repeatability of this method is good.

Table 1 The standard deviations of multiple measurement Unit :mm

Diameter of cylinder No.1	Diameter of cylinder No.2	Centre distance
0.000685	0.000522	0.001380

As shown in Table 2, because $122.5 - 0.0215 < 122.4980 < 122.5 + 0.0215$, the workpiece can go.

Table 2 The evaluation results of the rod bar Unit : mm

$D_{1m} + D_{2m}$	$D_1 + D_2$	T_1	ΔT	dc	GO/NOT GO
60.9660	60.9530	0.0215	0.0065	122.4980	GO

Conclusions

1. In the past, one size of comprehensive gage can only be used to evaluate one kind of workpiece. Since the manufacture of the gage is very expensive, it can not adapt to the flexible manufacturing system well. However, in this method the virtual gage can overcome the defect.
2. The manufacture error of the comprehensive gage will cause larger error to the final measurement result than that of the normal dimension gage does, while in our method, there is no manufacture error in the virtual gage simulated by the computer. So the use of this virtual gage instead of the comprehensive gage can reduce the error on a larger scale. The development of this method can avoid the misjudgement caused by the manufacturing error.
3. Further more, with the development of the computer technology and algorithm, it becomes a tendency to use the virtual gage instead of the material gage. Our method is just in line with this trend.

In a word, by using this method, the position error of the work piece can be tested and evaluated automatically and quantitatively .

A METHOD TO ENHANCE THE QUALITY AND RELIABILITY OF INSPECTION DATA FOR PRECISION MANUFACTURING APPLICATIONS

G. L. Lee, J. Mou
Arizona State University
Tempe, AZ 85287-5906

1. Abstract

Inspection is commonly used to scrutinize the quality of manufactured products against the established standards and specifications. However, the quality and reliability of many inspection processes are contaminated by various uncertainties. Two prominent sources for measurement uncertainties are: a) the imperfection of a measuring device and b) the dimensional deviation and geometric characteristics of a measured feature. Usually, the effect of both types of uncertainties is confounded with one and other. To assure the quality and reliability of any inspection process, measurement uncertainty needs to be addressed for all data acquisition activities. A method is also needed to identify and decouple the effect of confounded uncertainties. If this can be done, then the collected data can be properly adjusted and a more meaningful analysis results can be drawn. In this paper, the issues of uncertainty identification for machine calibration and dimension measurement using artifacts with various geometric features are discussed. Analytical models are derived to describe the confounded effect of both types of uncertainties. Finally, a case study is used to illustrate the confounded effect of the measurement uncertainties.

2. Introduction

Dimensional inspection is widely used by industry to determine whether the manufactured product meets the established standards and specifications of design and manufacturing. The accuracy and effectiveness of the inspection procedures and equipment are very important for precision manufacturing. Therefore, to ensure manufacturing products with high quality and reliability, measurement uncertainty needs to be addressed for all inspection data acquisition activities. A variety of techniques were developed to deal with the uncertainty in inspection, error analysis, and error compensation. However, those techniques were unsatisfactory for precision manufacturing applications due to the fact that the errors resulted from either imperfect measuring device or dimensional deviation of artifacts are usually neglected. For example, when workpieces are inspected, it is common to interpret the observed errors as the dimensional inaccuracy of the workpiece. This can be true only if the measuring devices are perfectly built. Otherwise, the inaccuracy of a measuring device will deteriorate the quality of the measurement (Mou and Liu 1992). On the other hand, when artifacts are used in machine calibration, it is common to interpret the observed errors as the positioning inaccuracy of the machine. This can be true only if the artifacts are perfectly made. Otherwise, any dimensional deviation of the artifact from the nominal dimension will deteriorate the effectiveness of the calibration and the derived error model.

Unfortunately, sources of error exist in any measuring system. There are the imperfect mechanical structures, the errors in control systems, and the environmental disturbances (Ferreira and Liu 1986). Meanwhile, it is very difficult if not impossible to make a perfect artifact. Therefore, uncertainty exists in both types of applications and their effect is usually confounded with one and other. Hence, it is necessary to develop a methodology to first identify and then decouple the measurement uncertainties that result from both machine related and artifact related errors. In this paper, a method is presented to discuss the issue of uncertainty identification in machine calibration and dimension measurement using artifacts with various geometric features. The measurement method and error classification for both the machine related and artifact geometry related uncertainties are discussed. Generalized models are derived to describe the confounded effect of both types of uncertainties. The derived models are validated for common measuring devices and measurement procedures. A case study is used to illustrate the confounded effect of the measurement uncertainties.

3. Analytical Models Derivation

In this study, we focus on the effect of machine error only and assume the artifact has ideal geometric features. Although dimensional error is a three-dimension problem, for simplicity, measurement uncertainty is discussed in two-dimension forms. Generally, basic surface features of ideal geometry include (part of) circles, rectangles, cones, and spheres. Their surface contour can be represented in two-dimension by two geometric elements: lines and arcs. For our purpose, the following basic geometric features are used to derive the analytical model for uncertainty identification and separation. They are: a) a skew cylinder or cone and b) a circle or sphere.

In most inspection operations, the data acquisition process is conducted while controlling each axis either individually or simultaneously. For all operation modes, the errors coming from all axes affect each other axis through the geometric feature of the artifact. When a designated nominal point $N = (x_N, y_N)$ on the workpiece is measured, the measuring process can be divided into two stages. In the first stage, the probe is quickly moved to a temporary point $N_I = (x_{NI}, y_{NI})$ which is a nominal point relative to N . Usually, for every measuring machine, the distance between N and N_I is the default approach distance d . From point N_I , the probe is then moved slowly toward the designated measurement point N in the second stage to assure an accurate measurement.

If there is machine error in both x-axis and y-axis, that is δ_x and δ_y , exist, the probe may not be exactly located at the point N_I , but will be located at P_I instead. Then, the probe will be moved along the designated approach direction that usually parallel to x-axis, y-axis, or normal to the feature surface until a contact occurs. However, due to machine related error, the machine controller cannot detect the error and will command the probe to move along a path slightly different from the designated path as shown in Figure 1. For simplicity, we assume the new probing path is parallel to the designated path with an offset. The offset is a function of machine errors. Depending on the relative measurement direction and geometric characteristics of the measured feature, the inspection data will be contaminated by different types of confounded uncertainties. The uncertainties are resulted from machine related errors and geometry related errors as shown in Figure 1 to Figure 6. In following sub-sections, the effect of machine related errors and geometry related errors is illustrated.

3.1 Measurement along the Y Axis

In this inspection process, the probe moves along the y axis while remain at the same nominal position in x axis. In this situation, the error in x axis is implicit and usually will not be detected by the machine controller. However, the error will affect the measurement accuracy along the y axis. The effect is confounded with the characteristics of the geometric features of a workpiece.

(a). Measuring a Skew Cylinder or Cone Feature

If the inclination angle (θ) of the geometric feature is know (see Figure 1), then $S = \tan(\theta)$. If machine errors exist in both axes, that is $\delta_x \neq 0$ and $\delta_y \neq 0$, the machine error is traditionally calculated as $\Delta y = \delta_y = y_N - y_{NI}$. However, we can clearly find that the distance between N and N_I should not be treated as the machine related errors here. It is not a direct error due to an imperfect machine, but an indirect error resulted from a confounded effect of machine error and geometric characteristics of the measured feature. By geometric analysis, the actual machine error in y axis should be expressed as $\delta_y = y_N - y_{NI} + \delta_x * S$. The relative measurement error is $\frac{\Delta y - \delta_y}{\delta_y} = \frac{-\delta_x * S}{\delta_y}$.

(b). Measuring a Circle or Sphere Feature

If the inclination angle (θ) of the tangent line at point N is know (see Figure 2), then $S = \tan(\theta)$. If machine errors exist in both axes, that is $\delta_x \neq 0$ and $\delta_y \neq 0$. In tradition, the error is calculated as $\Delta y = \delta_y = y_N - y_{NI}$. However, as similar to case (a), the machine related error in y axis should be expressed as

$\delta_y = y_N - y_{NI} + [\sqrt{R^2 - (X_N + \delta_x)^2} - \sqrt{R^2 - X_N^2}] = y_N - y_{NI} + [R \cos(\theta_2 - \frac{\pi}{2}) - R \cos(\theta - \frac{\pi}{2})]$. Therefore, the

$$\text{relative error is } \frac{\Delta y - \delta_y}{\delta_y} = \frac{-[R \cos(\theta_2 - \frac{\pi}{2}) - R \cos(\theta - \frac{\pi}{2})]}{\delta_y}$$

3.2 Measurement along the X Axis

In this inspection process, the probe moves along the x axis while remain at the same nominal position in y axis. In this situation, the error in y axis is implicit and usually will not be detected. However, the error will affect the measurement accuracy along the x axis and confounded with the characteristics of the geometric features.

(a). Measuring a Skew Cylinder or Cone Feature

If the inclination angle (θ) of the measured feature is know (see Figure 3), then $S = \tan(\theta)$. If machine error exists, the error is traditionally calculated as $\Delta x = \delta_x = x_N - x_{NI}$. However, the actual error in x axis should be

$$\delta_x = x_N - x_{NI} + \delta_y / S \text{ and the relative error is } \frac{\Delta x - \delta_x}{\delta_x} = \frac{-\delta_y / S}{\delta_x}$$

(b). Measuring a Circle or Sphere Feature

If the inclination angle (θ) of the tangent line at point N is known (see Figure 4), then $S = \tan(\theta)$. If machine error exists, the error is traditionally calculated as $\Delta x = \delta_x = x_N - x_{Nf}$. Nevertheless, the actual error in x axis should be

$$\delta_x = x_N - x_{Nf} + [\sqrt{R^2 - (y_N + \delta_y)^2} - \sqrt{R^2 - y_N^2}] = x_N - x_{Nf} + [R \sin(\theta_2 - \frac{\pi}{2}) - R \sin(\theta - \frac{\pi}{2})]. \quad \text{Therefore, the}$$

$$\text{relative error is } \frac{\Delta x - \delta_x}{\delta_x} = \frac{-[R \sin(\theta_2 - \frac{\pi}{2}) - R \sin(\theta - \frac{\pi}{2})]}{\delta_x}.$$

3.3 Measurement along the Normal Direction

In this inspection process, the probe moves along the direction that is normal to the ideal feature surface. In this situation, the machine errors will affect the measurement accuracy in both x and y axes. The effect is also confounded with the characteristics of the geometric features of a workpiece.

(a). Measuring a Skew Cylinder or Cone Feature

If the inclination angle (θ) of the measured feature is known (see Figure 5), then $S = \tan(\theta)$. The machine error is traditionally calculated as $\Delta x = \delta_x = x_N - x_{Nf}$ and $\Delta y = \delta_y = y_N - y_{Nf}$. However, the actual errors should be

$$\delta_x = x_N - x_{Nf} + \cos^2(\pi - \theta)\delta_x - \sin(\pi - \theta)\cos(\pi - \theta)\delta_y; \quad \delta_y = y_N - y_{Nf} - \sin(\pi - \theta)\cos(\pi - \theta)\delta_x + \sin^2(\pi - \theta)\delta_y.$$

$$\text{The relative errors are expressed as } \frac{\Delta x - \delta_x}{\delta_x} = \frac{-\cos^2(\pi - \theta)\delta_x + \sin(\pi - \theta)\cos(\pi - \theta)\delta_y}{\delta_x} \quad \text{and}$$

$$\frac{\Delta y - \delta_y}{\delta_y} = \frac{\sin(\pi - \theta)\cos(\pi - \theta)\delta_x - \sin^2(\pi - \theta)\delta_y}{\delta_y}.$$

(b). Measuring a Circle or Sphere Feature

If the inclination angle (θ) of the tangent line at point N is known (see Figure 6), then $S = \tan(\theta)$. If machine error exists in both axes, the error in x and y axes should be expressed as $\delta_x = x_{p2} - x_{pf}$ and $\delta_y = y_{p2} - y_{pf}$. The relative

$$\text{errors in } x \text{ and } y \text{ axes are } \frac{\Delta x - \delta_x}{\delta_x} = \frac{x_{p2} - x_{pf}}{\delta_x} \quad \text{and} \quad \frac{\Delta y - \delta_y}{\delta_y} = \frac{y_{p2} - y_{pf}}{\delta_y}.$$

$$\text{where } x_{p2} = R \cos(\theta - \frac{\pi}{2}) + \delta_x - \delta_y * \tan(\pi - \theta); \quad y_{p2} = R \sin(\theta - \frac{\pi}{2}); \quad y_{pf} = -\frac{x_{pf}}{S} + A;$$

$$x_{pf} = \frac{\frac{A}{S} + \sqrt{\frac{1}{S^2} A^2 - (1 + \frac{1}{S^2})(A^2 - R^2)}}{(1 + \frac{1}{S^2})}; \quad A = \frac{\delta_x}{S} + \delta_y$$

4. Simulation Analysis

Simulation analysis is conducted to illustrate the effect of the confounded errors. Let's assume the default value of the approach distance $d = 0.07874''$ (2 mm) and $\delta_x = \delta_y = 0.005''$. For both cone and sphere feature measurements, the confounded errors caused by machine errors and geometric characteristics of measured feature were shown in Figure 7. In both cases, the simulation was conducted for θ from 100° to 170° . The results show that the confounded effect of machine related and geometric related uncertainties may be large enough to significantly degrade the accuracy of any measurement. Then, all analytic results may be misled based on the measuring data without considering these confounded effects.

5. Conclusions

To ensure manufacturing products with high quality and reliability is a main goal in precision manufacturing. A variety of techniques were developed to deal with the uncertainty in inspection, error analysis, and error compensation. However, those techniques were unsatisfactory due to the fact that the errors resulted from either the measuring device or the deviation in workpiece dimensions is usually neglected. In this paper, an innovative method was presented to discuss the issues of uncertainty identification in machine calibration and dimension measurement

using artifacts with various geometric features. The measurement method and error classification for both the machine related uncertainty and workpiece geometry related uncertainty were discussed. Generalized analytical models were derived to describe the confounded effect of both types of uncertainties. Simulation analysis was then used to show that the errors resulted from both the measuring device and geometric characteristics of measured features can not be neglected and their effect is confounded with one and other.

6. Reference

Ferreira, P. M., and Liu, C. R., 1986, "An Analytical Quadratic Model for the Geometric Error of a Machine Tool," *Journal of Manufacturing Systems*, Vol. 5, No. 1, pp. 51-63.

Mou, J., and Liu, C. R., 1992, "A Method for Enhancing the Accuracy of CNC Machine Tools for On-machine Inspection," *Journal of Manufacturing Systems*, Vol. 11, No. 4, pp. 229-237.

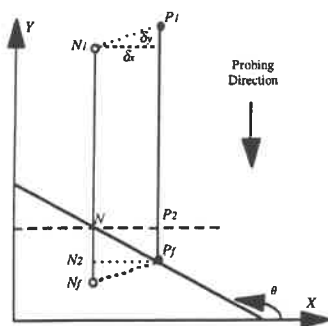


Figure 1. Measuring a Cone Feature along the y Axis

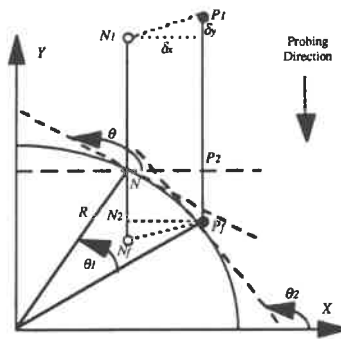


Figure 2. Measuring a Sphere Feature along the y Axis

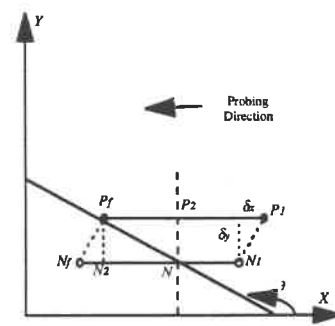


Figure 3. Measuring a Cone Feature along the x Axis

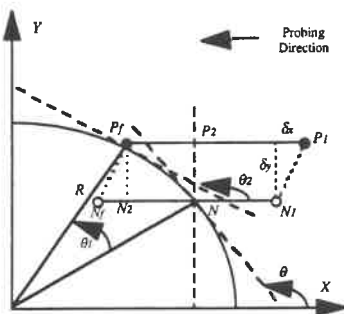


Figure 4. Measuring a Sphere Feature along the x Axis

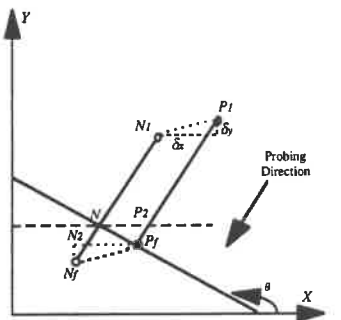


Figure 5. Measuring a Cone Feature along the Normal Direction

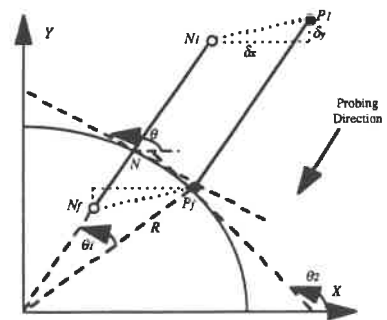


Figure 6. Measuring a Sphere Feature along the Normal Direction

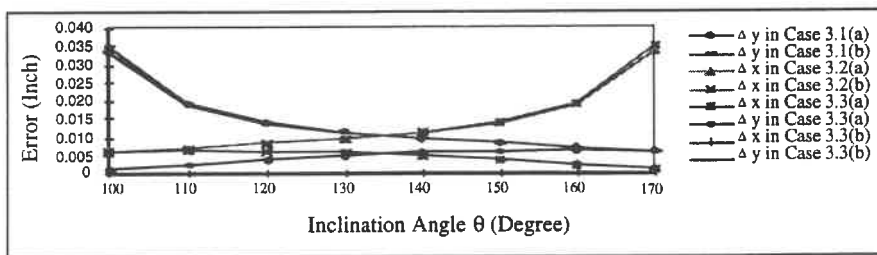


Figure 7. Confounded Errors Due to Machine Related Errors and Geometric Characteristics of Measured Features

Helical Screw Rotor Tolerancing and Coordinate Metrology for Grinding Process Optimization

Iwan H. Sutedjo, CQE, CQA, CMfgT
Heimir Fannar, PhD
James L. Bowman, PE

Ingersoll-Rand Company
Rotary-Reciprocating Compressor Division
Davidson, NC 28036

Robert G. Wilhelm, PhD

University of North Carolina at Charlotte
Precision Engineering Laboratory
Charlotte, NC 28223

This paper describes an approach for grinding process optimization that balances quality loss against inspection and manufacturing costs. The approach is applied to increase the quality level while minimizing the total costs in helical screw rotor manufacturing.

The helical screw compressor is a positive displacement compressor which functions by reducing the volume of an entrapped air pocket, and thus elevating its pressure. This entrapped volume is generated by meshing threads of two helical (twin) screw rotors which are enclosed by an outer housing. Due to relative velocity of the rotors, both with regard to each other and to the housing, some clearances are required in order for the compressor to function properly. These clearances must be controlled accurately in order to prevent excessive fluid leakage and the consequent loss of thermodynamic performance. Current manufacturing capability yields rotors that meet 15 micrometer clearance requirements; however, impending technology will require much tighter clearances to improve the compressor performance.

Twin screw compressors, which have in recent years become the most common type of compressors in industrial compressed air and gas applications, use pairs of helical screw rotors with very exacting requirements. Line contacts between the rotors are critical functional requirements that must be satisfied by controlling the profile condition on the rotor surfaces, and thus the entire surfaces must be accurately manufactured since every point on the surface will, at some time in the rotational cycle, form part of a sealing curve.

In the 'unsynchronized' screw compressor, a small amount of torque, typically amounting from 4% to 10% of the compressor driving torque, is transmitted from one rotor to the other. Considering this small torque transmission, one of the rotors is nearly idle in torsion. Ideally, this idle rotor should have a constant angular velocity with respect to the mating rotor. Any errors in the rotor's lead or index variation will cause cyclic fluctuations in the relative angular velocity between the rotors. These fluctuations cause noise and vibration in the compressor.

The rotor's functional requirements which determine the dynamic features of the compressor's rotating components depend upon these following features: profile, lead, and index variation. These features are defined by the established gear industry standard¹:

Profile is one side of a tooth in a cross section between the outside circle and the root circle. Usually a profile is the curve of intersection of a tooth surface and a plane or surface normal to the pitch surface, such as the transverse, normal, or axial plane.

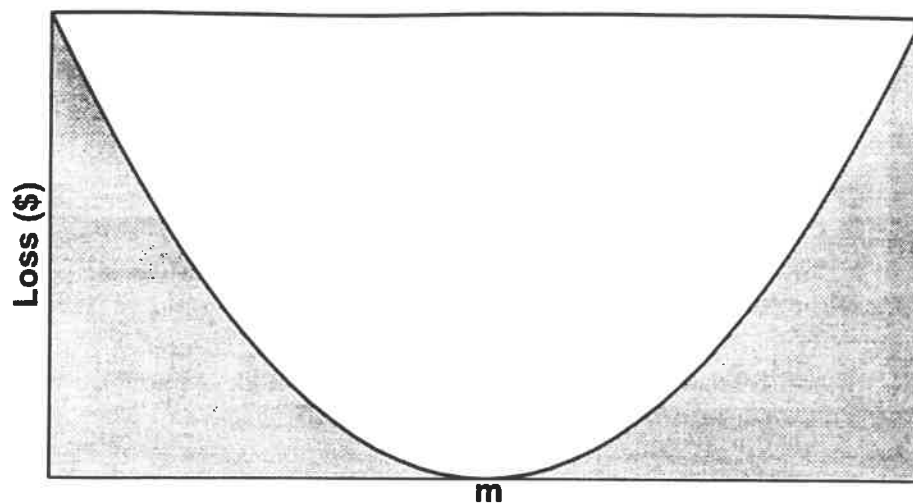
Lead is the axial advance of a helix for one revolution (360 degree), as in a screw thread, and the teeth of a helical gear.

Index Variation is the displacement of any tooth. Measurements are usually linear, near the middle of the tooth profile, and if made normal to the tooth surface, should be corrected to the transverse plane.

Since the twin screw rotors operate as pairs, the thermodynamic performance is determined by matching various features, such as lead, of the two rotors. For example, it is possible to have a constant lead error in one rotor while maintaining a high compressor performance provided that the lead error of one rotor is consistently matched by the corresponding lead error in the mating rotor. However, this manufacturing practice is proven costly because it requires selective pairing of components in the assembly process. Moreover, consistent rotor quality level is not maintained between batch to batch manufacturing.

To optimize the rotor grinding process, the Taguchi Quality Loss Function² method is used to minimize the inspection and manufacturing costs against the total quality loss. This technique requires quantitative measures of tolerance design and manufacturing performances. Furthermore, several cycles of tolerance design, manufacturing, and measurement are often necessary to arrive at an optimized process. Tolerance specifications, suited to helical screw rotors, are used to describe the design requirements and Coordinate Measuring Machine (CMM) results are used to quantify the manufacturing performance.

In order to utilize the Taguchi Quality Loss Function for improving quality, a standardized rotor inspection process is developed to provide a real-time feedback to the manufacturing process. The Taguchi Quality Loss Function shown in Figure 1 is based on the optimization of the monetary loss due to measurement accuracy and required feedback time to the manufacturing process. The rotor quality features which are considered for this inspection process are profile, lead, and index variation measurements.



$$\text{Loss (\$)} = k [\sigma^2 + (\mu - m)^2]$$

Figure 1: Taguchi Quality Loss Function

A four-factor with two-level balanced experiment is designed to establish a robust inspection process with a minimum required number of measurements. The measurement capability is then determined with identified controllable and uncontrollable (noise) factors. An example of this balanced experiment for a robust profile measurement is shown in Table 1.

Controllable Factors	Level 1	Level 2
A. Probe Diameter	3 mm	5 mm
B. Probing Distance	5 mm	8 mm
C. Profile Points Selection	3 points	7 points
D. Replication per Point	5 times	10 times

Table 1: Factors and Levels for 2^4 Profile Inspection Experiment

To analyze the above experiment with the highest resolution, i.e. all main factor effects and its interactions are estimated, the L_{16} Orthogonal Array (OA) with 16 trials is used for a balanced experiment. Then, an Analysis of Variance (ANOVA) is performed to identify control factor which effects the measurement mean. Also, a Signal to Noise (S/N) ratio for the Nominal is the Best (NB) condition is calculated to identify control factors which effect the measurement variation.

In the manufacturing setup process, the grinding wheel position setup is a critical issue for achieving the required rotor quality; therefore the setup conditions feedback, which is usually obtained at the inspection station (CMM), must be made in a real-time basis for immediate and accurate setup adjustment. It should be noted that during the inspection process, which may take up to 60 minutes (5 seconds per point on a touch-trigger probe CMM)³ depending upon the CMM speed and number of measurements, the machine tool remains idle until the interpreted inspection results are fed back to the machine tool for making appropriate setup adjustment prior to manufacturing runs.

It is obvious from the above that some parts must be manufactured in order to qualify the setup process. This qualification process requires immediate feedback with reliable data from the inspection process, i.e. the CMM, prior to starting the manufacturing process. In an optimized manufacturing process, the machine tool must be setup in the minimum amount of time required and with the minimum number of initial parts (some of which may have to be scrapped). The quality of the new machine tool setup is then determined by the first few parts inspection using the above robust inspection process.

It can be assumed that the grinding wheel's condition, e.g. its wear and consequently its remaining life, can be established from the previous manufacturing batch, i.e. from the last part inspection. Therefore, it is unnecessary to perform a complete grinding wheel wear or profile analysis during a new setup. However, the grinding wheel's wear and profile should be monitored throughout the manufacturing process with Statistical Process Control (SPC) by using the above proposed robust inspection process. The proper use of SPC can easily predict the grinding wheel wear rate and thus its remaining life.

The practice described above ensures optimum manufacturing process with manufacturing process interruption done only when a machine tool adjustment is deemed necessary by the SPC charts evaluation, and not because of waiting for an interpreted inspection result feedback. Thus, the manufacturing process optimization is hereupon defined as a time function of the corresponding inspection process feedback.

References:

1. ANSI/AGMA 2000-A88, *Gear Classification and Inspection Handbook*, American Gear Manufacturers Association, 1988.
2. Ross, P. J., *Taguchi Techniques for Quality Engineering: Loss Function, Orthogonal Experiments, Parameter and Tolerance Design*, McGraw-Hill, New York, 1988.
3. Mestre, M. and Abou-Kandil, H., "Measuring the Errors of Form of Industrial Surfaces: Prediction and Optimization", *Precision Engineering*, 16 (4): 268-275, 1994.

DESIGN AND ANALYSIS OF A TWO-DIMENSIONAL ALUMINUM STANDARD TO CALIBRATE CMMs

Padilla, S., Palomino, D., Ascanio G., Ruiz, G. and Sánchez, J.
Centre of Instruments, UNAM, P. O. Box: 70-186, 04510 México, D.F.

INTRODUCTION

As we need to have a standard for the acceptance, verification and calibration tests of Coordinate Measuring Machines (CMM), we carried out a study of a prototype object which could join several features such as: capability to satisfy the three tests mentioned above, and besides be a simple object, easy-to-handle, with suitable and adequate dimensions for a variety of CMM, easy-to-transport, easy construction and calibration, as well as good stability and moderate price. With all of these features kept in mind its creation should be possible using the present facilities at the Centre of Instruments (CI) belonging to The National University Autonomous of Mexico (UNAM).

DEVELOPMENT

For this plan we established three steps: design, manufacturing and calibration. The design step has two stages: the first for the analysis of different alternatives and founding the deformations caused by an object's shape and dimensions; the second stage for analysis of deformations produced by fittings and tolerances in machining, moreover the arising of deformations due to temperature variation and its corrections. In this paper we describe the results obtained from the first stage of the first step of the plan.

Analyzing the metrology characteristic qualities of several objects developed as standards for CMM^[1], we look for an object that offers both, the qualities mentioned above and simultaneously the most of useful information nevertheless the lack of resources (i. e., computer) of the CMM under test. At the beginning, we proposed a two-dimensional object consisting of a rectangular metal plate as prototype of the reference standard, which contains an orthogonal array formed by the center of geometric bodies to allow that set up and performing of tests could be achieved in a short time, and easy understanding of results, with the center-to-center distance as the length for testing. Considering that these geometric bodies should be measurable in any CMM, we selected spheres, however, to obtain the optimum size and quantity of measurable objects with the largest measurable area, were selected spherical-cavity insets.

As the material for the plate it was selected aluminum thinking in low weight to meet the easy-to-handle limitation^[2] imposed. To determine the size of the plate, several tests were carried out laying out a new limit: deformation (only mechanical strength) $< 2 \mu\text{m}$ from center-to-center to keep reliability on the object as standard, nevertheless the position in the measuring volume of CMM. Accordingly to analysis results, the dimensions and shape of the plate were modified, moreover the size of the insets, based on deformations produced by three free-supports, which position also was modified searching the best fitting with minimum deformation.

The first prototype was proposed as a rectangular plate (600 mm x 800 mm) of aluminum 7075T6 and 51 mm of thickness, containing 32 drills (52 mm diameter) into an orthogonal array (rows and columns equally separated to 100 mm). For the insets the properties of steel SAE-H13 were considered. The position of the center of each cavity will be calibrated to know its coordinates (x, y, and z) related to a reference (an edge of the prototype), and the distance between those coordinates will be the reference length for test; that is why we need that such centers stay in its position with the shortest displacement as possible during storage, transportation and testing. For the test, the standard should be placed in horizontal and vertical positions, keeping reliability on accuracy for each center-to-center distance, for any position of the plate, specially for horizontal position, because it is necessary lay down the plate onto adjustable supports for leveling. For such purpose, was determined a three-free-support system, and the corresponding static analysis of the plate to observe the resulting deformations. Those analyses and all the last analyses were carried out numerically by mean of finite element analysis (FEA) using NISA-DISPLAY^[3] software.

RESULTS

The earlier analysis for the plate was performed in two dimensions before drilling, the second analysis (2D) including the 32 bores without spherical cavities, and the third analysis (3D) including bores and cavities. The maximum deformation found was $3.937 \times 10^{-2} \mu\text{m}$ and maximum center-to-center deformation around 12 μm , pointing out that the rigidity decreases with insets installed, exceeding the imposed limitations for calibration purposes. As the running of 3D analysis took much time, the following analyses were run for 2D models, and only to achieve the minimum displacements required, the 3D analysis was carried out.

To obtain from FEA reliable results, 3D analysis was carried out running software in several trials until get results independent of the mesh size to determine the minimum of elements to construct the model, assuming that starting from this minimum the result can be considered as reliable, and looking for invariability if that number of elements is increased. In this way, the 3D models were constructed using 8,000; 12,000 and 25,000 first-order elements for the plate with a combination of quadrilateral and triangular elements, and finally was selected a model with 25,000 elements because the results obtained for the displacements shown maximum errors less than 7%.

So, the following software running was made to determine the optimum thickness of the plate keeping the maximum displacements between center of spherical-cavity insets were less than the 2 μm stated (into the range of the minimum reading for many CMM), finding a thickness of 51 mm; however, with these dimensions, the plate is very heavy for handling and transportation purposes (65 kg approximately) consuming long time to set up the calibration.

Taking as a basis the mentioned above, several models were proposed using a plate of 51 mm thickness, adding drills to loss weight and designing a new support system with more points of contact, however, it was noted that rigidity decreases as the number of drills increases, and it decreases even more inserting the spherical cavities, resulting in an increasing of deformations, as we don't get a significant reduction of weight, because the weight of removed aluminum is almost equal to weight of spherical cavities.

For this reason, the thickness was changed to 25 mm, using new dimensions for the plate, and a different amount of drills, considering in this new case: 550 mm x 700 mm with 63 drills; and 325 mm x 475 mm with 24 drills. Drills of 47 mm diameter equally spaced to 75 mm from center-to-center. Starting from 2D analysis, to seek the optimum position for support pads for the minimum deformations, resulting 31 positions for the first plate and 5 for the second one, performing later equal number of software runnings as a number of positions found. The maximum deformation for both plates starts from 5.4 μm up to 21.4 μm for the large plate and starts from 1.5 μm up to 4.3 μm for the small plate. So, the best position for the supports, for the small plate results in a deformation in length, within the range of 2 μm stated as shown in figure 1 and its maximum deformation is 1.5 μm .

The 3D analysis result for the small plate is shown in figure 2 where the maximum error is 1.47 μm using red color and the maximum deformation for distance center-to-center of spherical cavities is less than the stated 2 μm .

CONCLUSION

By this way, even though both plates meet with suitable dimensions and reference lengths for calibration purposes, assuming as possible the minimum deformation, finally, we decide the manufacturing of the small plate because two basic reasons: weight (proportional to size) and cost. However, we must carry out new analyses for this plate to determine the complexity of computation required to correct the deformations caused by temperature variations during measuring process.

REFERENCES

- [1] Hegelman, R., Trapet, E., and Wäldele, F., *Coordinate Metrology*, Tutorial of the 2nd International Conference on Ultra-Precision in Manufacturing Engineering, May 1991. Germany.
- [2] Padilla, S., Palomino, D., Ascanio G., Ruiz, G., Nava, R., and Sánchez, J., *Diseño de un Patrón de Verificación para MMC*, VIII Congreso de Instrumentación, October 1993, Xalapa, México.
- [3] E. M. R. C., DISPLAY III, NISA II, DISPLAY POST, "User's manual", Engineering Mechanics Research Corporation, 1991, Michigan, U. S. A.

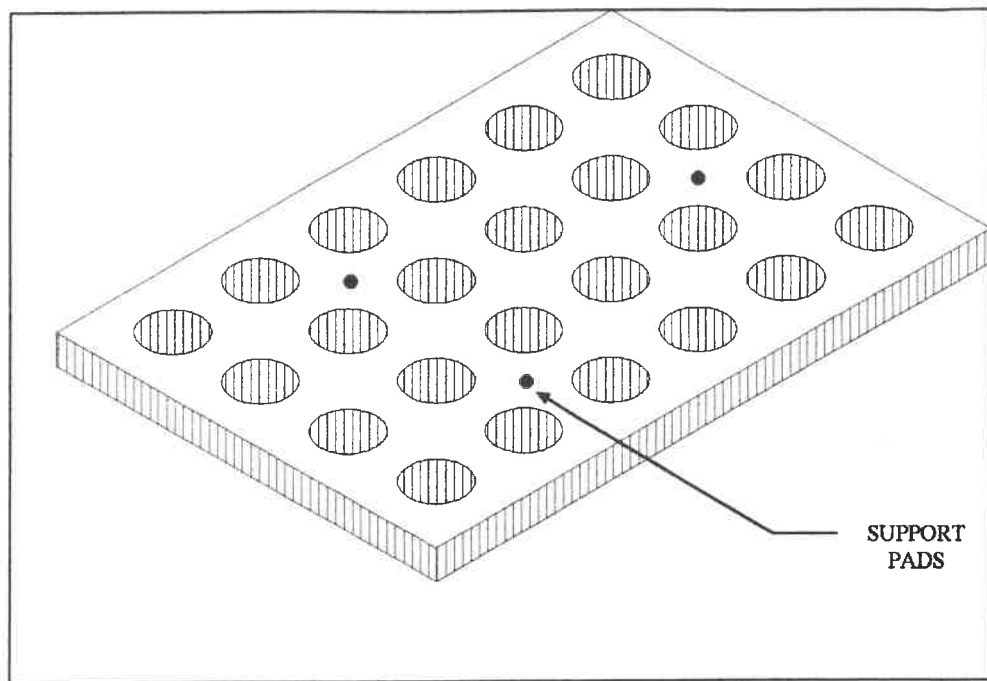


Fig. 1. Scheme of the plate and support pads

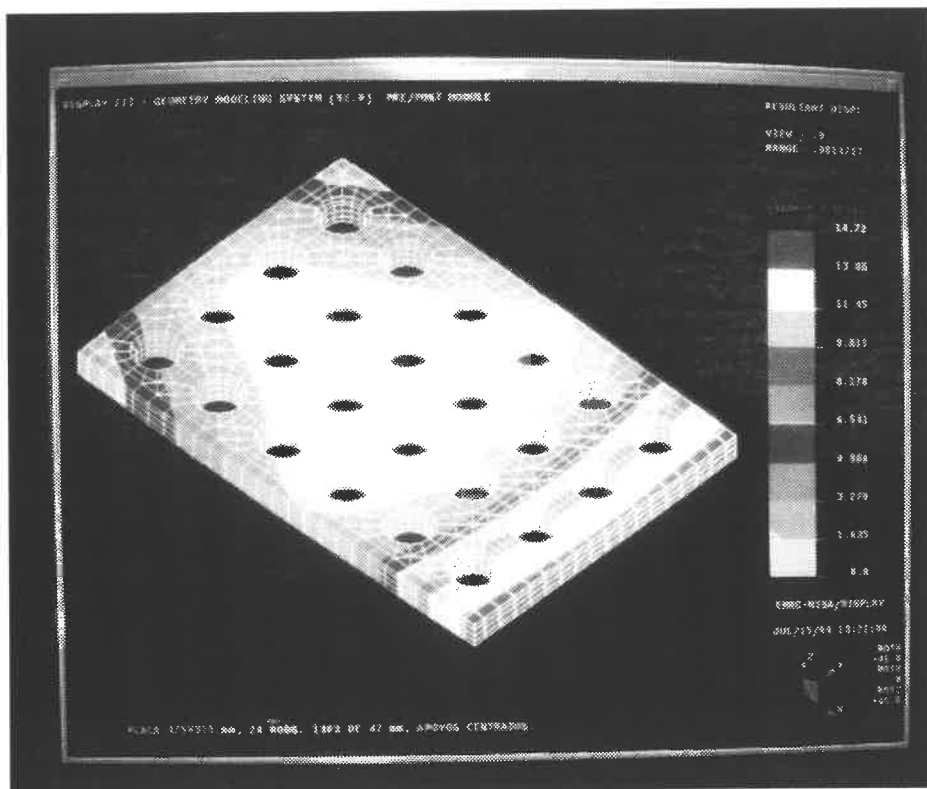


Fig. 2 Minimum deformation obtained with FEA

Generalization and Verification of Vectorial Tolerances

Hong-Tzong Yau

Department of Mechanical Engineering
National Chung Cheng University

Introduction

Recently, the integration of coordinate measuring machines (CMMs) and CAD systems has promoted a new approach to the evaluation of workpiece geometry, namely, Vectorial Dimensioning and Tolerancing. This new approach is promising in that it defines dimensions and tolerances that are process related in a clear and distinct way. Therefore it enables proper manufacturing control and process diagnosis [1,2]. Interests have been generated on both sides of Atlantic that some suggests vectorial tolerancing be used as an alternative means for representing workpiece tolerances in complement to the conventional geometric tolerances (defined by ISO 1101 and ANSI Y14.5). In Europe, coordinate measurement methods utilizing vectorial tolerancing principle will soon be published in the ISO TR 10 360-1. The principle of vectorial dimensioning and tolerancing is based on the assessment of substitute elements. Substitute elements are geometrically ideal features that were derived from the measurement points of the actual workpiece surfaces. As the substitute element is derived, four descriptive parameters, namely position, direction, size, and form, are used to represent the geometry that is being assessed. Tolerances are allowable deviations being assigned to the first three of the four for representing vectorial tolerances. Figure 1 shows the position vector and direction vector in a workpiece coordinate system.

Shortcoming of Current Vectorial Tolerancing System

One of the major differences between vectorial tolerancing and conventional tolerancing is that vectorial tolerancing separates the location and orientation from the feature parameters and treat them as general parameters for all the substitute elements. The advantage is that it facilitates the generalization of defining and evaluating substitute elements independent of their element types.

However, by definition the direction vector which specifies orientation is a unit vector. In order to avoid over-specifying, only two of the three components of the substitute direction vectors can be tolerated. In other words, although the direction vector has three components, it actually controls only two degrees of freedom. Each component is always constrained by the other two, making it redundant in terms of specifying orientations and tolerances. When the nominal direction vector is not parallel to any axis of the substitute datum system, it becomes confusing as to how to assign tolerances to the vector components. In fact, 3D orientations are under-specified using direction vectors. The use of direction vectors makes it difficult to represent and evaluate true 3D geometric elements such as free form surfaces.

A New Orientation Vector

Due to the deficiency of the current direction vector for specifying orientation and correlating with functions, we need a true 3D orientation vector to establish proper

relationship between substitute elements and substitute datum. In the fields of robotics and kinematics, people have been using rotation angles and rotation matrix for the representation of orientation for a long time. Although conceptually, rotation angles are a little more difficult to visualize than a direction vector, but functionally they can represent a true 3D orientation without any ambiguity. In this work, it is proposed that the Z-Y-X Euler angles $A = [\gamma, \beta, \alpha]^T$ [3] be used to describe the orientation relationship between substitute elements and substitute datum. A rotation matrix can then be written as

$${}^D R = \begin{bmatrix} c\alpha c\beta & c\alpha s\beta s\gamma - s\alpha c\gamma & c\alpha s\beta c\gamma + s\alpha s\gamma \\ s\alpha c\beta & s\alpha s\beta s\gamma + c\alpha c\gamma & s\alpha s\beta c\gamma - c\alpha s\gamma \\ -s\beta & c\beta s\gamma & c\beta c\gamma \end{bmatrix} \quad (1)$$

where c and s are abbreviations for cosine and sine functions respectively. In fact, the last column of the matrix is equivalent to the old direction vector. A transformation of coordinates from the substitute element coordinate system to the substitute datum coordinate system can be written as

$${}^D Q = {}^D R {}^S Q \quad (2)$$

where $Q = [x, y, z]^T$. Together with the position vector P_0 , a general location and orientation transformation can be represented as:

$${}^D Q = {}^D R {}^S Q + {}^D P_0 \quad (3)$$

Using homogeneous transformation, a more compact form can be obtained:

$${}^D Q = {}^D T {}^S Q \quad (4)$$

where $T = \begin{bmatrix} R & P_0 \\ 0 & 1 \end{bmatrix}$ and $Q = [x, y, z, 1]^T$.

Formulation

With the tools introduced in the previous section, we are now ready to formulate and generalize vectorial tolerances. A point on a substitute element viewed from the datum can be defined as

$${}^D S_j = {}^D T {}^S N_j \quad (5)$$

where ${}^S N_j$ is the corresponding point on the substitute element viewed from the local coordinate system of the substitute element and

$${}^D T = {}^D T(R(A + \Delta A), P_0 + \Delta P_0) \quad (6)$$

and

$${}^S N_j = {}^S N_j(d + \Delta d, u, v) \quad (7)$$

where d is the size parameter(s) and u, v are the parametric variables of the surface. A local form deviation can thus be represented as

$$\varepsilon_j = ({}^D M_j - {}^D S_j)_{\min} \quad (8)$$

where ${}^D M_j$ is a measurement point taken by a DME (Dimensional Measurement Equipment). Notice that the subscript indicates that the form deviation is normal to the surface of the substitute element. Therefore, for a given transformation and a fixed size parameter(s), we can obtain ${}^D S_j = {}^D T^S N_j(\bar{d}, u_0, v_0)$ by solving the following set of equations:

$$\frac{\partial \varepsilon_j}{\partial u} = \frac{\partial \varepsilon_j}{\partial v} = 0 \quad (9)$$

Finally by definition, the total form deviation is

$$Form = (\varepsilon_j)_{\max} - (\varepsilon_j)_{\min} \quad (10)$$

Evaluation

Based on the least squares (or Gaussian) principle, an objective function can be set up to evaluate the vectorial tolerances:

$$F = \sum_j \varepsilon_j^2 = \sum_j ({}^D M_j - {}^D T^S N_j)^2_{\min} \quad (11)$$

The best fit solution can be found by minimizing F with respect to A , P , and d .

$$\frac{\partial F}{\partial A} = \frac{\partial F}{\partial P_0} = \frac{\partial F}{\partial d} = 0 \quad (12)$$

The solution for the parameters can be found by solving the nonlinear equation set (11). The nominal position and rotation vectors and sizes can be used as initial values for the solution. The difference between the solutions and the nominal values are the deviations that can be compared with permissible tolerances. Although nonlinear in nature, there exist robust and reliable methods for solutions. A unified nonlinear least squares algorithm has been developed to solve equations in the format of equation (11) [4].

Notice that the formulation does not indicate any particular type of substitute element. In this sense, the formulation is a general approach to evaluating various types of substitute elements, including free form surfaces. Moreover, it is also applicable to fitting multiple surfaces. To achieve that, equation (11) can be extended for multiple surfaces with the subscript i representing a particular surface.

$$F = \sum_i \sum_j \varepsilon_{ij}^2 = \sum_i \sum_j ({}^D M_{ij} - {}^D T^S N_{ij})^2_{\min} \quad (13)$$

This is important for application such as pattern hole fitting and workpiece alignment in NC machining and CMM inspection.

Measurement of Turbine Wheel Die Segment

Figure 2 shows the measurement analysis of a turbine wheel die segment. The surfaces in the model were generated using B-Spline surfaces. The two side-by-side contour surfaces control the function and performance of the turbine wheel surfaces. A scanning CMM was programmed to scan both of these functional surfaces after initial part alignment has been made. Before best fit, the peak-to-valley form error is at the magnitude of 0.18 mm and the standard deviation is 0.036 mm. After the simultaneous fit

of these two functional surfaces, the total form error is reduced to 0.11 mm and the standard deviation 0.016 mm. The total number of measurement points is about 1200. The evaluated position and rotation vectors represent the shift or offset of the die segment surfaces to the substitute datum. This is the error created in the machining set-up. Using the evaluated location and orientation deviations, this set-up error can be corrected or compensated by straight-forward coordinate transformation.

Conclusions

This paper proposes a new rotation vector in vectorial tolerancing for representing true 3D orientations. It is proved that the rotation vector overcomes the problem of under-specifying the degrees of freedom for 3D orientation. With the old direction vector, only symmetric geometric elements such as planes, cylinders, and cones can be represented. With the new rotation vector, free form surfaces can also be represented.

The rotation vector and position vector can be combined to form the homogeneous coordinate transformation. Similar to various applications in Kinematics and Robotics, it provides a sound basis for the mathematical definition and evaluation of vectorial tolerances. The representation is generic in that it applies to general 3D surfaces. As a result, multiple surfaces can be fit simultaneously. This property lends itself not only to pattern hole fitting but also to workpiece alignment that is typical in NC machining and CMM inspection.

References

- [1] Wirtz, A., 1991, "Vectorial Tolerancing for Production Quality Control and Functional Analysis in Design", CIRP International Working Seminar on Computer-Aided Tolerancing, pp. 77-84.
- [2] Henzold, G., 1993, "Comparison of Vectorial Tolerancing and Conventional Tolerancing", Proceedings of the 1993 International Forum on Dimensional Tolerancing and Metrology.
- [3] Craig, J., 1986, "Introduction to Robotics", Addison-Wesley publishing company.
- [4] Yau, H.T. and Menq, C.H., 1992, "A Unified Least Squares Approach to the Evaluation of Geometric Errors", ASME Winter Annual Meeting, PED-Vol. 56.

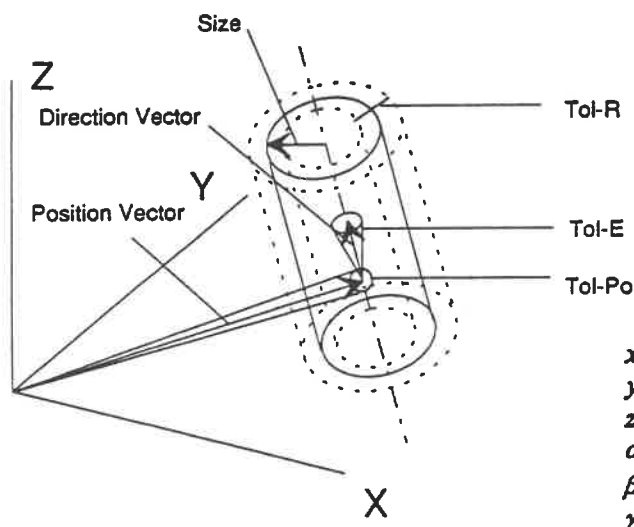


Figure 1 Illustration of vectorial tolerancing

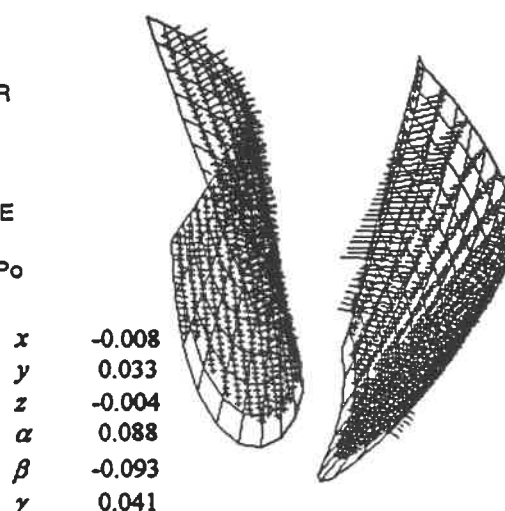


Figure 2 Evaluation of die segment surfaces.

Optical Probe for Surface Profiling of Aspherical Mirrors

P. S. Huang and Xiaorong Xu
Department of Mechanical Engineering
SUNY Stony Brook
Stony Brook, NY 11794-2300

1 Introduction

Surface profiling of large aspherical mirrors such as grazing incidence x-ray mirrors is usually difficult using conventional full-aperture interferometric profiling systems. For this reason, several scanning profiling systems have been developed, including systems based on interferometry and laser autocollimation. In these systems, surface slope rather than surface height is measured. Surface profile is then obtained through numerical integration of the measured surface slope. One representative of such profiling systems is the Long Trace Profiler (LTP) developed by Takacs *et al.* at Brookhaven National Lab [1] which is based on the pencil beam interferometer concept. The profiler can measure surface profiles over spatial periods ranging from 2 mm to 1 meter with nanometer order resolution. LTP has been successfully used for surface profiling of various aspherical mirrors for synchrotron radiation applications.

In this paper, we describe a new slope measuring optical probe for surface profiling applications. The probe is based on a new angle measurement method developed by Huang *et al.* [2-4] which makes use of the internal reflection effect of light beams. The advantages of this probe lies in its simple principle, high resolution, and low cost. The probe is especially effective for measuring surface profiles of mirrors with large radius of curvature.

2 Principle

Figure 1 shows the optical layout of the probe. A single-mode-fiber-coupled laser diode (wavelength 675.4 nm) is used as the light source because of its compactness and high beam quality. With a pigtailed collimator, the laser source provides a well collimated beam with approximately 2 mm of diameter and 1.2 mW of output power. The collimated beam is then directed to a polarized beam splitter (PBS) where its *s*- and *p*-polarized components are separated. The *p*-polarized component, which is not used here, passes through the PBS and is absorbed by a piece of black sponge. The *s*-polarized component of the beam is reflected at

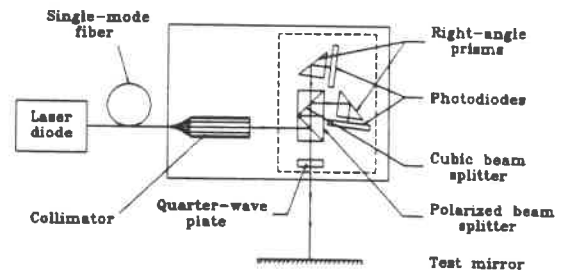


Figure 1 Optical layout of the probe.

the PBS and is then further reflected by the test mirror. A quarter-wave plate is placed in between the PBS and the test mirror so that the light beam returning to the PBS will change its polarization state from *s* to *p* and therefore will pass through the PBS. The laser beam passing through the PBS is divided into two beams with equal intensity by a cube beam splitter. These two beams are then each reflected by a right-angle prism and are finally each collected by a photodiode that measures the beam intensity. The photo current signals from the photodiodes are amplified and conditioned by an on-board electronic circuit. The processed signals are then sent to a PC via a 16-bit data acquisition system. The output signal of this probe is

$$S = K \left(\frac{I_1 - I_2}{I_1 + I_2} \right)$$

where I_1 and I_2 are the beam intensities measured by photodiodes 1 and 2 respectively and K is the amplification of the circuit.

Figure 2 shows the cutout view of the probe. Figure 3 is a photo of the probe. The dimensions of the probe are approximately 80 x 60 x 40 mm.

3 Experiment

3.1 Calibration

The calibration of the probe is performed by using HP5529A laser calibrator as the reference instrument. HP5529A can measure angular

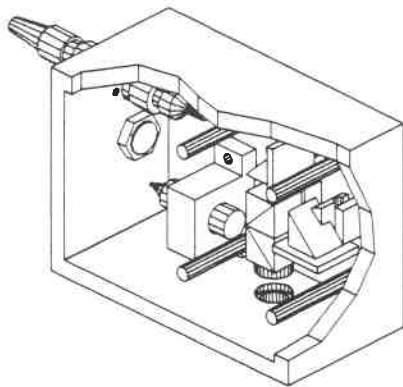


Figure 2 Cutout view of the probe.

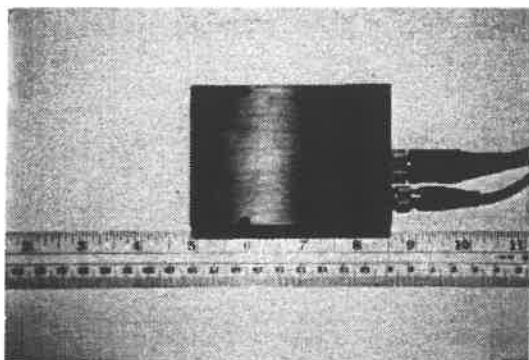


Figure 3 Photo of the probe.

displacement with a repeatability of 0.1 arcsec and an accuracy of $\pm 0.2\%$. To check the repeatability, the calibration was done twice. Figure 4 shows the results in the range of -1000 arcsec to 1000 arcsec. From linear fitting, we found the nonlinearity error to be approximately $\pm 0.25\%$ which is much larger compared to the theoretical results. This discrepancy is mainly caused by the existence of ghost beams. In the future design, wedged optical components or anti-reflection coatings should be effectively used to control or reduce the ghost beams so that the nonlinearity error can be further reduced.

In our experiment, we used curve fitting to reduce the effect of nonlinearity error. This was possible because of good repeatability of the calibration results (the difference between the first and the second result is less than 0.02%). A seventh order polynomial has been used, which resulted in a residual error of less than $\pm 0.04\%$.

3.2 Stability Tests

Both long-term and short-term stability tests were performed. The long-term stability test was

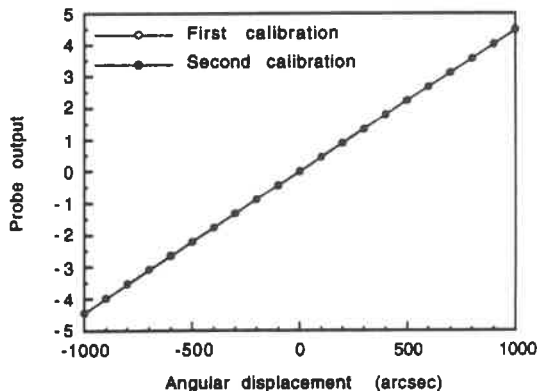


Figure 4 Calibration results.

carried out over a period of 12 hours with a data sampling rate of one point per 2 minutes. The result is shown in Figure 5. From this figure, we see that the probe drift in 12 hours is about 1.2 arcsec. Figure 6 shows the result of the short-term stability test. The test started shortly after the laser power was switched on. The experiment lasted for 16 minutes with a sampling rate of one point per 2 seconds. From this figure, we can see that the probe drift in the first 12 minutes is relatively large (0.65 arcsec). After that, the output is stabilized to its noise level which is approximately 0.1 arcsec peak-to-peak. This phenomenon can be explained by the unstable performance of the laser source at the warming-up stage. From these results, we can say that the optical probe has an angular resolution of 0.1 arcsec. The corresponding height resolution depends on the sampling step. For a sampling step of 2 mm (the step used in our experiment), the height resolution should be approximately 1 nm. This value can be further reduced by decreasing the sampling step.

3.3 Surface Profile Measurement

The test mirror used in our experiment is a concave spherical mirror with a nominal radius of curvature of 10 meters. Figure 7 shows the experimental setup for measuring the profile of the mirror. The scanning motion is provided by a coordinate measuring machine (CMM). The test mirror is mounted vertically on a mounting plate held by the vertical ram of the CMM. The CMM steps horizontally (step size 2 mm) so that the mirror can be scanned and sampled. The yaw error of the CMM during the scanning motion is monitored by HP5529A laser calibrator and then removed from the measurement results. Once the surface slope data are collected, numerical integration is performed to find the surface

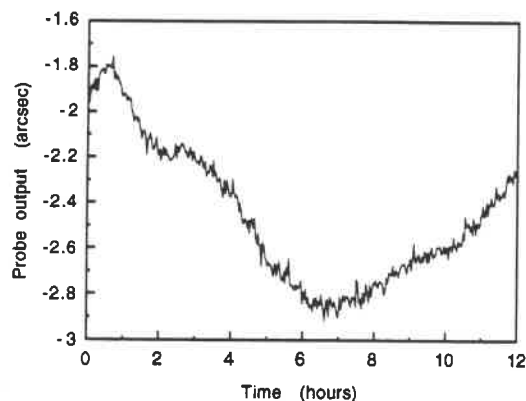


Figure 5 Long-term drift of the probe.

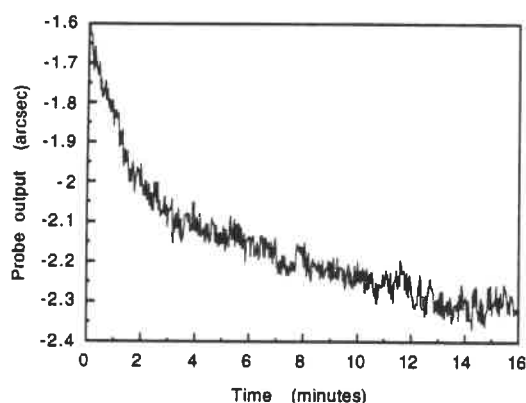


Figure 6 Short-term drift of the probe.

profile. The same experiment was done twice to check the repeatability of the measurement results.

Figure 8 shows the yaw error of the CMM, which varies from -0.5 arcsec to $+0.5$ arcsec during the scan. The results of two scans show large difference. This means that the yaw error of the CMM is not repeatable at this level. Therefore, for high accuracy measurement, the yaw error should be monitored in real time and then removed from the slope data.

Figure 9 shows the measured surface slope of the mirror along the scanned line after the yaw error of the CMM has been subtracted. Figure 10 shows the detrended surface slope with the linear component removed. The peak-to-peak difference between the two scans is approximately 0.8 arcsec. Figure 11 displays the surface profile of the mirror obtained from numerical integration of the slope data. The first order component is removed from the profile data because it represents the initial tilt of the test mirror and therefore has no significance here. From this figure, we found that the sag of

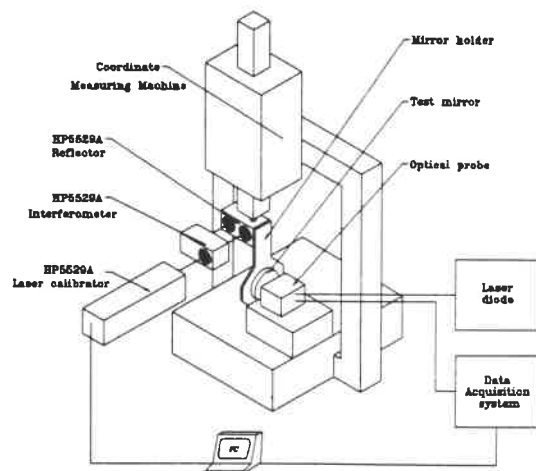


Figure 7 Experimental setup for profile measurement of mirrors.

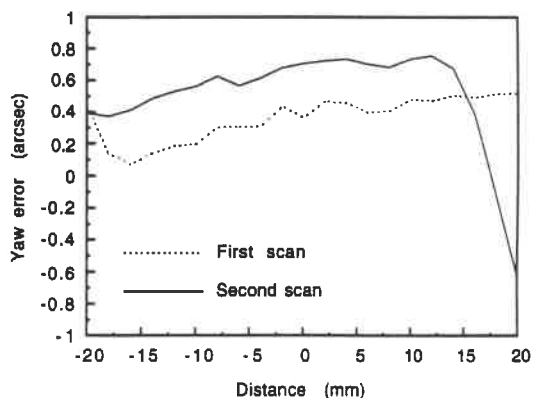


Figure 8 Yaw error of the CMM.

the mirror is $20.118 \mu\text{m}$, which corresponds to a radius of curvature of 9.941 meters. Figure 12 shows the detrended surface profile or the residual surface profile after the average radius of curvature has been removed. The repeatability of two scans is better than 3 nm .

4 Conclusion

We have developed a new optical probe for surface profile measurement of mirrors. The principle is based on a recently developed method for angle measurement, namely, Angle Measurement based on the Internal Reflection Effect (AMIRE). Surface profiles of mirrors are obtained through numerical integration of the local slopes of the successive points along the scanned line. Experimental results show that the angular resolution of the probe is approximately

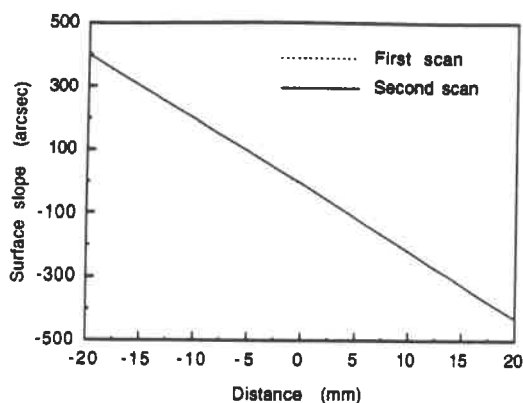


Figure 9 Surface slope of the mirror.

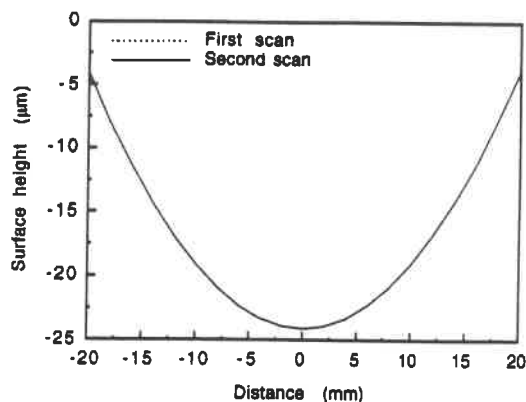


Figure 11 Surface profile of the mirror.

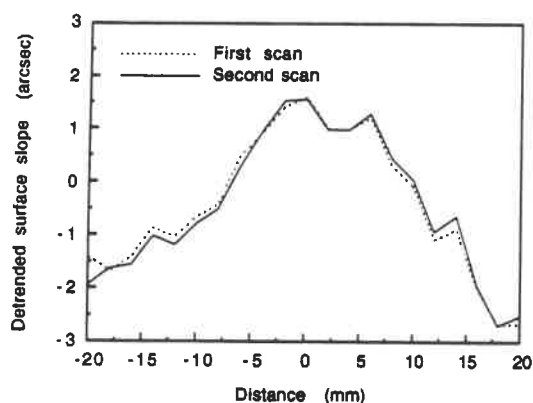


Figure 10 Detrended surface slope.

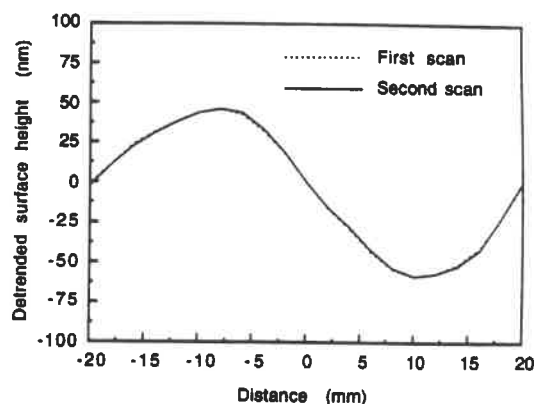


Figure 12 Detrended surface profile.

0.1 arcsec, which translates into a height resolution of 1 nm if the step size is 2 mm. In addition to its high resolution, the probe also has the features of compact size (80 x 60 x 40 mm) and low cost.

Both the resolution or the measurement range can be further improved in the future. For almost flat mirrors, the resolution can be further increased by either electronically amplifying the signal or using the multiple-reflection technique [4]. For mirrors with smaller radius of curvature, measurement range can be increased by using *s*-polarized beam, with some possible sacrifice in resolution.

Acknowledgment

The authors thank Dr. Peter Z. Takacs of Brookhaven National Laboratory for providing the test mirror used in this research as well as for his encouragement and helpful comments.

References

- [1] Peter Z. Takacs, Shi-nan Qian, etc., "Design of a long trace surface profiler", SPIE Vol. 749, Metrology, 59 - 64 (1987).
- [2] P. S. Huang, S. Kiyono, and O. Kamada, "Angle measurement based on the internal-reflection effect: a new method", Applied Optics, Vol. 31, No. 28, 6047-6055 (1992).
- [3] P. S. Huang and J. Ni, "Angle measurement based on the internal-reflection effect and the use of right-angle prisms", Applied Optics (in print).
- [4] P. S. Huang and J. Ni, "Angle measurement based on the internal-reflection effect using elongated critical-angle prisms", Applied Optics (in print).

NONCONTACT SURFACE ROUGHNESS MEASURING SYSTEM BASED ON COMPUTER VISION

You Zheng Chen Jun Pu Xinglin
Dept. of precision instruments
Tsinghua University, Beijing 10084 P.R.China

1. introduction

Assessing the quality of surface roughness is a very common problem in manufacture engineering. Most workpieces are evaluated by the inspection of some surfaces, and most workpieces contain numerous interfaces between surfaces. In many occasions, qualitative methods for evaluating the surface are employed. In others, quantitative measurements are required. In these cases, many surface roughness measurements are completed by using various methods. the most commonly used are stylus instruments, light section and interference.

The stylus instruments contact surface with stylus, risking surface damage. Optical interference instruments are noncontact instruments depending on fine adjusting. They have high accuracy, sensitivity and wide measuring range, but measuring instruments based on these methods are complicated, expensive, requiring high qualified operators to operate, susceptible to environmental disturbance and unfortunately their measuring speeds are considerably slow.

Modern production cycles requires fast procedures for roughness control, which are able to check several workpieces per minute. Traditional mechanical techniques are enable in doing so , moreover the other methods mentioned, if used within these speed constraints, are disadvantageous. So a new type profilometer must be carried out to obtain quantitative surface profile like stylus technique, but has high measuring speed mentioned above and without contacting the sample.

In this paper, a new surface roughness measuring instrument is described. A 2mW power laser source, a cylindrical lens and a CCD camera with suitable optical system are used to measure surface profile of samples. A real time video digitizer card writes the digitized image of the profile into the memory by processing date, so microprofile of the surface is shown on the screen and the evaluating parameters of the surface roughness such as Ra are got.

2. principle of the measurement

The equipment employed by the authors in measuring the roughness values is light section, microscope or double tube microscope. The principle of optical system is shown in Fig.1.

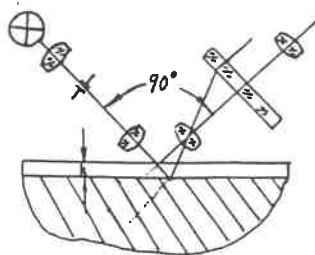


Fig.1 The principle of optical system

The light source system can focus the light into a fine beam 5mm in length and 0.2mm in width, the microscope system can make the light beam illuminate on the surface of sample, and finally make the measured image on the optic-electric detector---CCD camera.

The measured height of valley or peak of the surface is h :

$$h = \frac{N}{k} \cos 45^\circ$$

N ---the height of valley or peak of the surface in CCD

k ---the magnification of microscope system

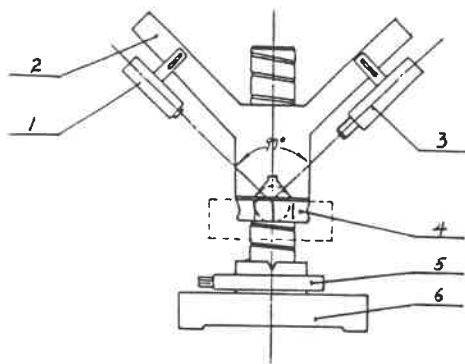


Fig.2 is the experimental setup

1 -- laser, 2 -- support part, 3 -- CCD camera, 4 -- adjust part

5 -- working table, 6 -- base

3. measuring result and analysis

The measured image in the CCD camera can be digitized by image processing card and the digitized profile which is shown in Fig.5,

And then this digitized profile is smoothed by the software. We get the profile of the measured image as shown in Fig.6.

At last, by filtering dashed area, the measured profile of surface roughness is got which is shown in Fig.7.



Fig.5 Digitized test profile

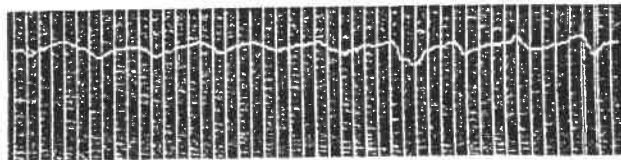


Fig.6 The profile of the smoothed image

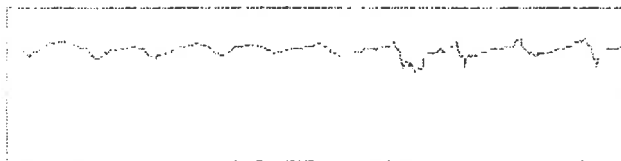


Fig.7 The surface profile of measured surface

Table below shows some results for surface roughness.

Unit:um			
No.	sample length	standard value of the grooved standard sample	measure value
1	0.25mm	0.2	0.182
2	0.8mm	0.4	0.398
3	0.5mm	4.2	4.185

4. conclusion

A new technique for roughness which is based on computer vision is discussed. The range of surface roughness value R_a is from 0.2 ~ 10 μ m. The arithmetical deviation is 10nm. Error of indication is less than $\pm 5\%$. Variation of indication is less than 4% and the stability of indication is less than 2% within 4 hours.

Reference

- [1] White field R.J Noncontact optical profilometer App.Opt.
1985.10 pp2480-2485
- [2] You Zheng etc. Noncontact optical heterodyne profilometry
Metrology acta 1993 No.2 pp20-25

A COMPARATIVE STUDY OF SURFACE TEXTURE MEASUREMENT USING WHITE LIGHT SCANNING INTERFEROMETRY AND CONTACT STYLUS TECHNIQUES

B.D. Boudreau, J. Raja, and H. Sannareddy
Precision Engineering Laboratory
UNC-Charlotte

P.J. Caber
WYKO Corporation

1.0 INTRODUCTION

Surface texture generated by metal removal processes has been measured using stylus type instruments for well over half a century. In recent years, a number of optical techniques have been developed for measuring surface texture. Instruments based on these techniques were initially developed to measure optically smooth surfaces. These instruments overcame some of the disadvantages of stylus type techniques in that they are non-contact and can quickly provide three-dimensional assessment of surface texture. However, instruments based on phase measuring interferometry were limited to very smooth surfaces and small fields of view. Recently, the capability of these instruments has been extended to the measurement of rough surfaces or step heights using white light vertical scanning interferometry. This extension has been achieved through increases in the vertical range and field of view. Commercial instruments are currently available with a vertical range of 500 μ m and field of view up to 10 x 10mm. The enhanced range and improved field of view have now made this type of instrument very useful in the numerous industries.

The improved capabilities also make these instruments appropriate for characterizing surfaces produced by metal removal processes. The measurement of these surfaces has traditionally been the domain of stylus instruments. This paper presents the results from a comparative study of the profiles obtained from a scanning white light interferometer and the profiles obtained from a stylus instrument.

2.0 INSTRUMENT CHARACTERISTICS

The surface texture measurements in this study were performed using two different instruments. The optical instrument was a WYKO Roughness/Step Tester (RST) while the stylus instrument was a Form Talysurf (FTS). The characteristics of each instrument are described in the following sections.

WYKO Roughness/Step Tester (RST):

The RST is a vertically scanning interference microscope system that operates with one of several interchangeable magnification objectives. Each objective contains an interferometer, consisting of a reference mirror and beam splitter, that produces interference fringes when light reflected off the reference mirror recombines with light reflected off the sample. When short-coherence white light is used, these interference fringes are present only over a very shallow depth on the surface. The surface is profiled by scanning vertically so that each point on the surface produces an interference signal and then locating the exact

vertical position where each signal reaches its maximum amplitude.

Form Talysurf S5:

The stylus instrument used in this study has a 4mm vertical range with 10nm resolution, and 120mm horizontal scan range. A conical stylus with a 1.5-2.5 μ m nose radius and 90° included angle was used. In general, the finite dimensions of the stylus mechanically filters very small surface wavelengths. The characteristics of this filtering depend on the radius and the included angle of the stylus. In addition, if a skid is used in the measurement it results in additional mechanical filtering of the longer wavelengths. The measured surface profile is then filtered to separate roughness and waviness.

3.0 RESULTS

The instruments described above were used to measure a number of different surfaces. The profiles from these measurements were analyzed in detail to understand the correlation between the two measurement techniques. The validity of this comparison requires careful consideration of the measurement procedure. The measurement procedure should ensure that the profiles are collected from the same location with similar ordinate spacing. This has been achieved by first physically profiling the same location using both instruments to obtain two profiles. The stylus profile ordinate spacing was then adjusted to be the same as the optical profile's using an iterative down sampling procedure. The resulting profiles were then quantitatively compared. The ensuing sections describe the measurement and analysis of these profiles.

Measurements were performed on a variety of surfaces. These surfaces ranged from reference specimen from NIST (sine wave) and PTB (repeatedly random ground surface), and manufactured surfaces (e.g. vertical milling and turning). These surfaces were chosen to cover a wide range of surface roughnesses and lay patterns. The results for the PTB specimen are discussed below.

Random Profile Calibration Standard:

Examination of random profile calibration standards allows for the evaluation of the measurement interaction over a wide range of wavelengths. The random profile calibration standards are a set of ground surfaces that have a random surface profile that repeats at regular intervals. The PTB reference surface used in this study had an average roughness (R_a) of 0.54 μ m, maximum peak-to-valley length (R_z) of 3.51 μ m and repeat interval of 2mm. The analysis results for this surface are given in figures 1 and 2. Figure 1 shows only minor differences between the down sampled stylus profile and the RST profile. The power spectrum plots show differences only in the short wavelength region which corresponds with the deviations observed in the profiles.

4.0 CONCLUSIONS

Surface texture measurements have been done using white light scanning interferometry and stylus profilometry on a variety of surfaces. In order to compare the profiles from the two instruments, measurements were carried out in the same region and then matched using correlation techniques. The results of the comparison have shown that white light scanning interferometry is capable of measuring machined surfaces.

Profile ID	R_a (μm)	R_q (μm)	R_t (μm)
FTS Original	0.5353	0.6709	3.6414
FTS Down Sampled	0.5290	0.6666	3.4128
RST	0.5008	0.6226	3.1548

Table 1 - Parameter values for PTB reference specimen.

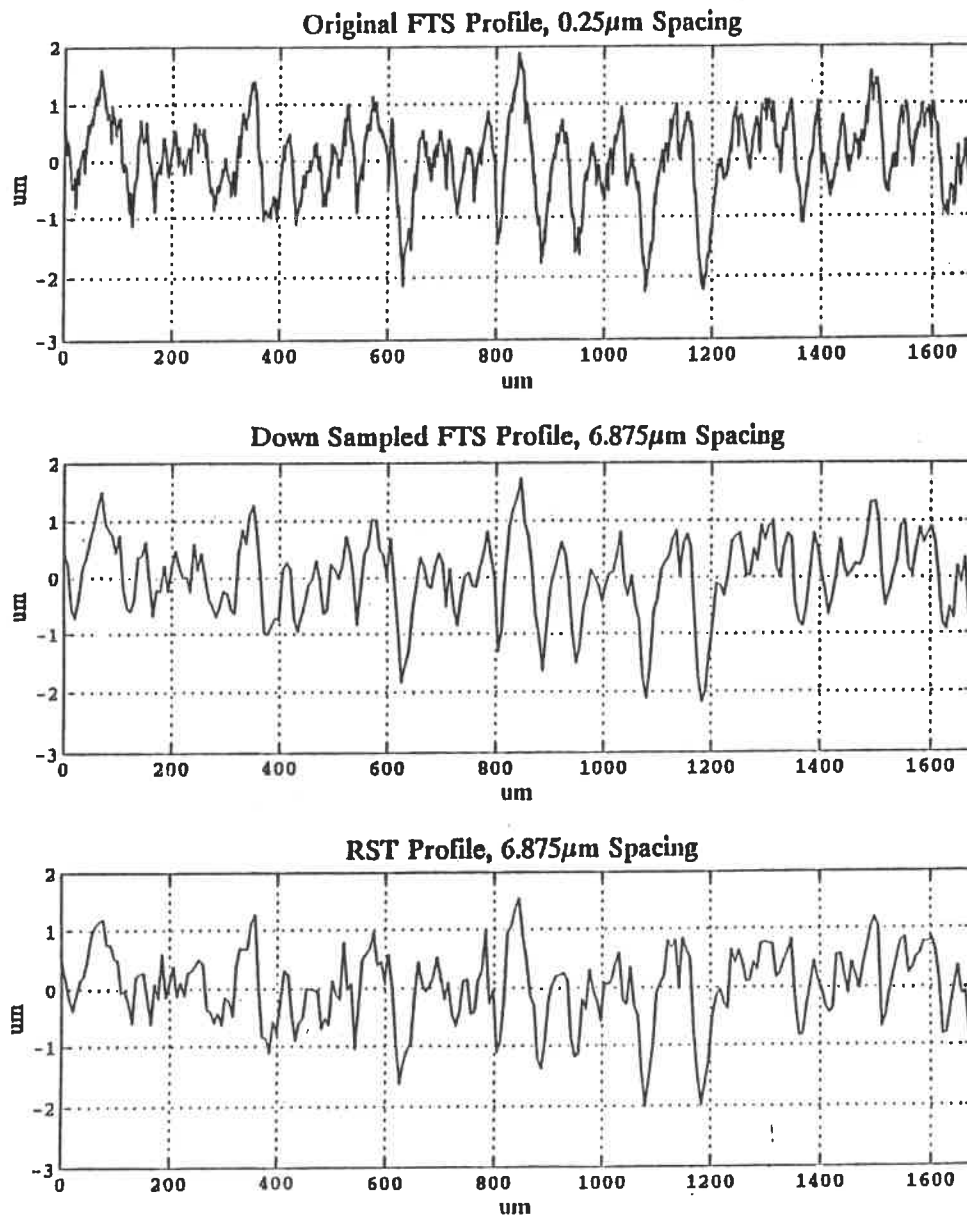


Figure 1. Unfiltered surface profiles from PTB roughness specimen

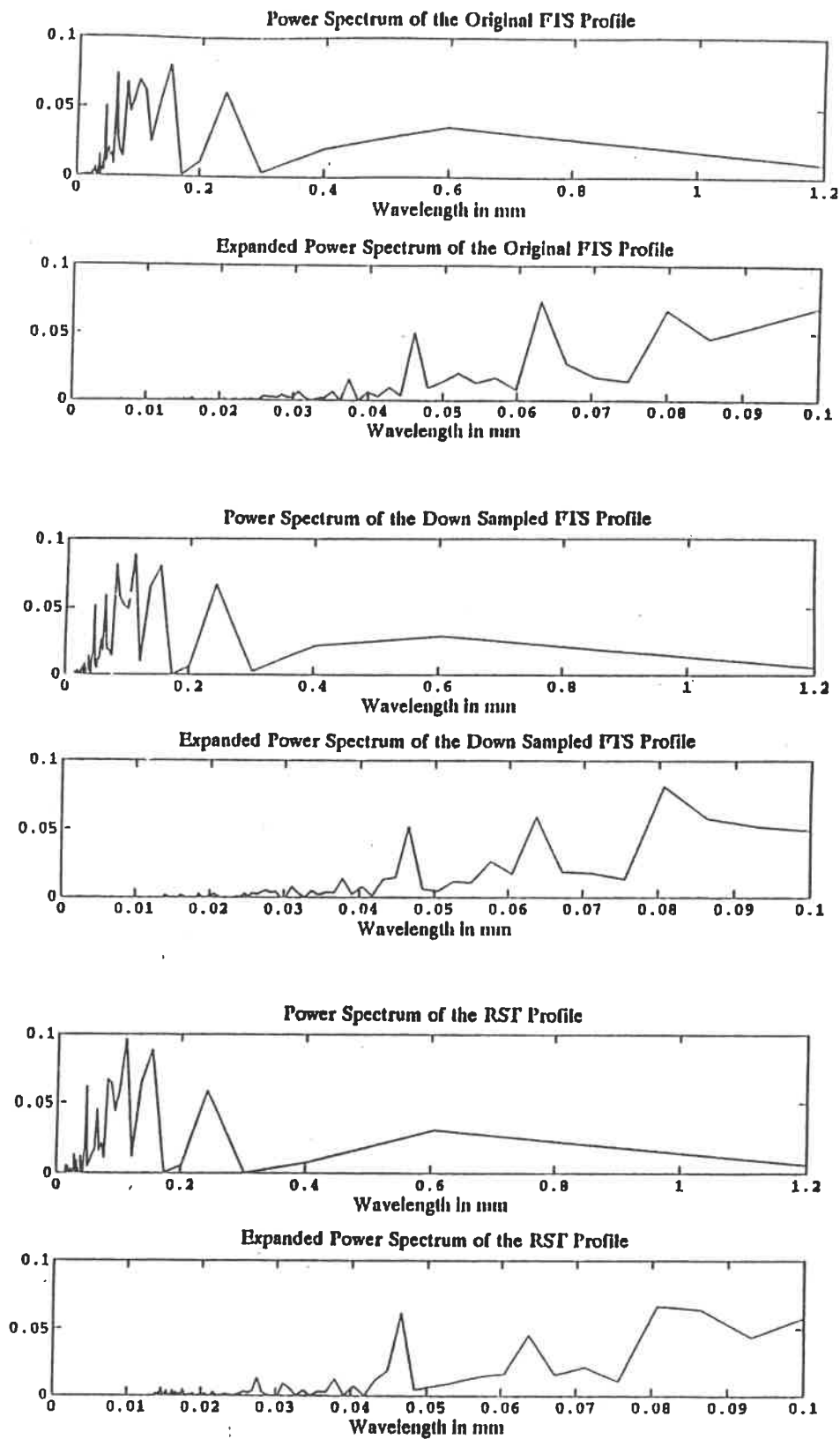


Figure 2. Power spectrum plots of profiles from PTB specimen

A Study of the Surface Microtopography of Aluminium Alloys in Single Point Diamond Turning

S. To and W.B. Lee

Department of Manufacturing Engineering, The Hong Kong Polytechnic University
Hung Hom, Kowloon, Hong Kong

ABSTRACT

Aluminium alloys are used as substrate materials in the manufacture of hardisk which requires high quality surface finish. Orthogonal cutting tests were performed on an aluminium alloy with a single point diamond tool under various machining conditions to study the microtopography of the machined surface. A high surface finish was obtained at a cutting depth of $3\mu\text{m}$ and at a feed rate between 20-30 mm/min. It is found that higher order statistical calculations such as the power density spectrum and autocovariance are important parameters in the study of ultraprecision machined surface.

INTRODUCTION

Aluminium alloys are used as substrate materials in the manufacture of hardisk. The amount of information that can be packed on the magnetic medium is related to the quality of the coating which also depends on the surface finish of the substrate material. A superfinish of the substrate is required for the subsequent defect-free coating of the magnetic film.

Ultraprecision machining based on single point diamond turning (SPDT) has become virtually an essential process in the design and manufacturing of high technology products such as memory disc substrates, precision mould inserts, aspheric lens and mirrors, and optic-electronic products[1,2,3], due to their stringent requirements in surface finish. The surface microtopography that can be achieved in a machined workpiece reveals a lot of information about the changes in the machine tool, cutting tools and workpiece conditions. Despite the wide application of the theory on random process analysis, meaningful correlation between the surface topography and the manufacturing process has yet to be established. The conventional stylus method is not suitable for the measurement of such ultraprecision machined surface as the stylus force can sometimes damage the surface. Phase-shifting interferometry provides a reliable method of sampling and obtaining geometrical information of microsurface roughness with high resolution and repeatability. The measurement can be performed at high speed.

In this paper, the effect of feed rate and cutting depth on the microsurface roughness of an aluminium alloy under various machining conditions are studied. Some statistical parameters used in the characterization of the diamond turned surface are described. Machining was performed on a two axis CNC contouring lathe using a single point diamond tool. The surface microtopography was measured with phase-shifting interferometric microscope (model WYKO-TOPO3D).

EXPERIMENTAL CONDITIONS

A two axis CNC contouring lathe from Rank Pneumo was used for the cutting experiments. The machine utilizes precision dovetail slides and high speed air bearing spindle with an integral AC drive motor. The slides are mounted on a natural granite. Spindle rotation is infinitely variable from 5000 to 10000 RPM with acceleration and deceleration times of less than 5 seconds. The workpiece is mounted in a air actuated collet mechanism within the air bearing spindle. Single point diamond tools was used in the experiment. The radius of the cutting edge of the diamond tools is $1.5\text{ }\mu\text{m}$ and the rake angle is 0° .

The workpiece materials used in the experiment are aluminium alloy 6061 in small discs of 12.7 millimeter diameter. All cuts were made at a spindle speed of 8000 rpm without lubrication. The depth of cuts used in single point diamond turning was much smaller compared with those used in conventional machining and ranged from 1 to 5 micrometer, whereas the feed rate used was from 10 to 50 millimeter per minute.

EFFECT OF MACHINING PARAMETERS

Figure 1 shows the effect of different feed rates on the microsurface roughness of the diamond turned surface at a depth of cut of $3\text{ }\mu\text{m}$. Experimental results shows that when the feed rate was increased initially from 10 mm/min to 30 mm/min, the surface roughness improved slightly. However, when the feed rate increased beyond 30 mm/min, the surface roughness (RA) rapidly increased. Fine mirror surface was obtained at a feed rate between 20-30 mm/min. The effect of cutting depth at constant feed rate on the RA value of the machined alloy is shown in Figure 2. The roughness value does not increase or decrease monotonically with cutting depth but shows a somewhat cyclic pattern with the lowest surface roughness value at a depth of cut between 2-3 μm . No build-up edge was detected at this depth of cut. Further investigation will be needed to explain such erratic results.

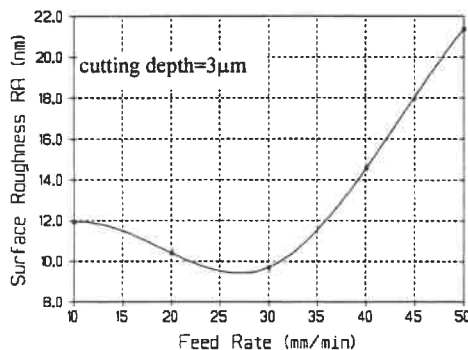


Figure 1 Effect of feed rate on cutting surface roughness.

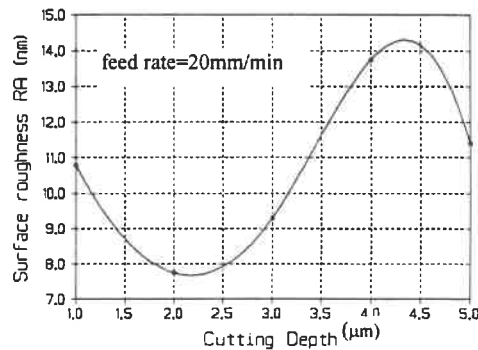


Figure 2 Effect of cutting depth on cutting surface roughness.

MEASUREMENT OF MICROTOPOGRAPHY

(a) The interferometric technique

Various implementations of interferometry have been used to measure the microsurface roughness diamond turned surface[4,5]. The optical data used to construct a topography of sample surface are created from a phase interferometer. The paths of light reflected from the workpiece surface and the path of light reflected from the reference surface were combined inside the magnification head to produce fringes. Multiple frames of the interferogram are used to determine the optical path difference (OPD) between the test and reference, which are then used to calculate the surface height difference. The interferometer and computer are connected by an electronic interface unit, which digitize the information from the interferometer. The data are then transferred to a computer and the surface height deviations are calculated. Modern interferometric microscopes are equipped with powerful computer software and various statistical parameters can be computed and displayed instantly.

From the digital phase-shift interferometry microscope, the first order statistical parameters such as the average height (RA), root mean square height (rms), and the higher order statistical parameters such as the slope, curvature, power spectrum and autocovariance function can be obtained [6].

(b) Measurement results

A three dimensional surface plot of the machined alloy at a cutting depth of $3\mu\text{m}$ and a feed rate of 20 mm/min are shown in Figure 3. Different heights are shown by different shades of color or grey level. The area being sampled was 3 mm by 3 mm and the rms is 9.51 nm . A two dimensional surface profile plot of the same surface is shown in Figure 4. The curve is a point by point display of the relative height of the surface being examined. The power density spectrum (PDS) plot of the diamond turned surface is shown in Figure 5 for four orientation (0° , 45° , 90° , -45°) on the surface. The power spectrum represents a display of the total power at each spatial frequency displayed on a logarithmic scale, which is obtained by squaring of the Fourier Transform of the surface height. The PDS plot for the four orientation are closed to each other and remains fairly constant.

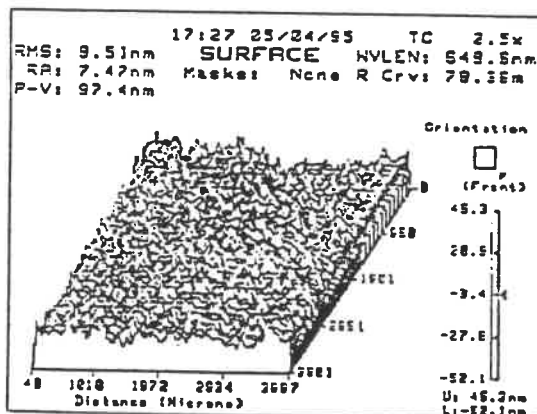


Figure 3 A three dimensional plot of a diamond turned aluminium substrate.

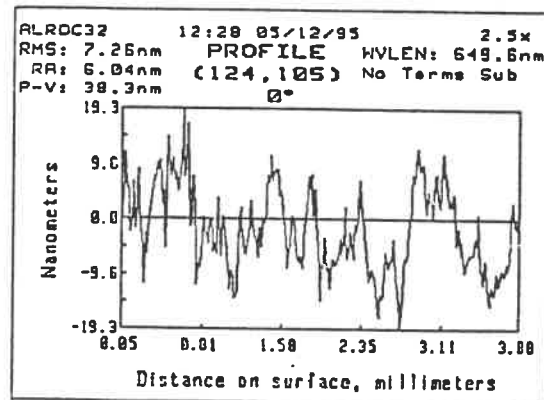


Figure 4 Two dimensional surface profile plot of diamond turned aluminium substrate.

An autocovariance plot shows the correlation length and enhances periodic features that exist on the surface. The precision diamond turned substrate shows fine periodic features in all orientation (0° , 45° , 90° , -45°) on the surface as shown in the autocovariance function of Figure 6. The autocovariance function enables random features of the machined surface to be separated from the periodic components of the surface.

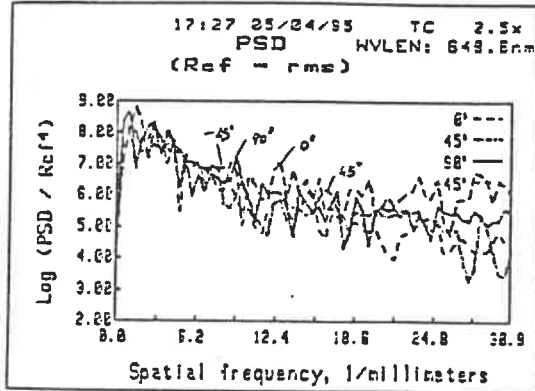


Figure 5 A power spectrum plot of diamond turned aluminium substrate

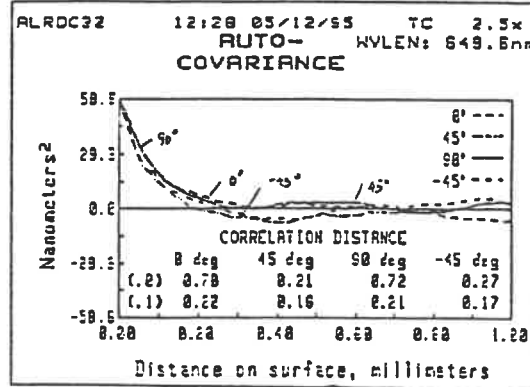


Figure 6 A autocovariance profile plot of diamond turned aluminium substrate.

CONCLUSION

In ultraprecision machining with single point diamond tool, a proper choice of the feed rate and cutting depth is important to optimize the surface finish. Good surface finish was obtained at a feed rate of 20-30 mm/min and at a cutting depth of 2-3 μm .

The powerful software of modern interferometric microscopes enables statistical parameters to be computed rapidly from the surface microtopography. The autocovariance function and the power spectrum, both of high order statistics, are particularly useful to characterize diamond turned surfaces.

Further work is being planned on the proper choice of the high order statistical parameters which will be useful to help diagnose surface deviations caused by individual effects such as diamond tool wear, grain anisotropy and material inhomogeneity.

REFERENCES

- [1] Strohmeier, B.R., Evans, W.T. and Schrall, D.M., "Preparation and surface characterization of zincated aluminium memory-disc substrate", Journal of Materials Science, 28, 1993, p.1563-1572.
- [2] Sugano, T., Takeuchi, K. and Yoshida, Y., "Diamond turning of an aluminium alloy for mirror", Annals of the CIRP Vol. 36, 1, 1987, p.17-20.
- [3] Yuan, Z.J., Zhang, M., He, J.C., Han, R.J., and Ma, W.S., "Surface integrity in ultra-precision machining with SPDT", Proceedings of the 6th IMCC, Hong Kong, May 1993, p. 27-36.
- [4] Jacob, P., "Optical measurements of diamond-turned surfaces", SPIE Vol. 1038 Sixth Meeting in Israel on Optical Engineering, 1988, p.260-264.
- [5] Azzazy, M., "Surface topography diffuse point differential interferometry", Optics and Lasers in Engineering, Vol. 10, 1989, p.81-96.
- [6] Whitehouse, D.J., "Handbook of surface metrology", Institute of Physics Publishing Bristol and Philadelphia, 1994, p.49-72.

FEEDBACK CONTROL OF A LOW-PROFILE MICRO-POSITIONING STAGE

Ping Ge, M. Jouaneh
Department of Mechanical Engineering
University of Rhode Island
Kingston, RI 02881

Introduction

In high-precision joining processes such as laser welding, there is a need for a positioning stage that have a large motion range and a nanometer resolution. *Figure 1* shows a schematic of the piezo-driven, low profile, vertical motion flexure-type micro-positioning stage that was designed for this application. The

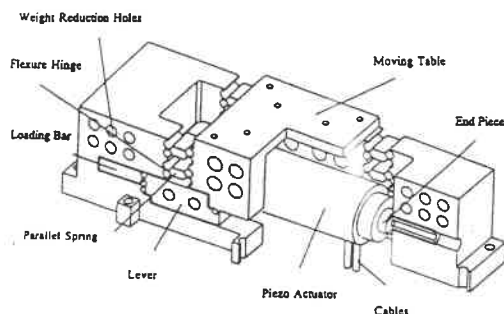


Figure 1. The Fabricated Micro-Positioning Stage

performance of the micro-positioning stage was reported as having a vertical motion range of 200 μm , high stiffness, and high positioning accuracy.¹ Tests on the micro-positioning stage were carried out using the built-in strain gage position sensor on the piezo actuator as a feedback sensor. No work was performed on studying the behavior of the stage when mounted on two Klinger stages which add x - y coordinates of motion. To obtain good dynamic performance, the control strategy should consider the dynamics of the overall system. This requires that the feedback control signal be derived from the absolute displacement of the combined stage system rather than from the built-in position sensor on the piezo actuator. This paper discusses how a digital PID controller with a

lowpass filter improves the performance of this micro-positioning tower with multi-vibration modes.

Dynamic Modeling

The micro-positioning stage was designed as a combination of a piezo-actuator, a flexure-hinge guided moving table, and a flexure-pivoted lever arms to amplify and transfer the horizontal expansion of the actuator into vertical displacement of the stage. In the design of this stage, each flexure hinge was placed to act as a "torsional spring". Thus, the vertical displacement of the stage is the result of the rotational motion of these hinges about a horizontal axis. Because of symmetry, the micro-positioning stage can be simplified as an equivalent mass and spring system with a single DOF along the vertical direction. Since the micro-positioning stage sits on an x - y Klinger stage which is considered as an additional mass-

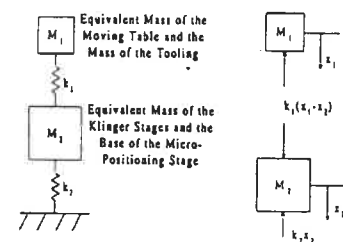


Figure 2. Vibration Model of the Combined Micro-Positioning Stage System

spring system, the combined system becomes a two DOF system. Thus, a dynamic model of the combined system can be built up as a coupled vibration system (see *Figure 2*). If we neglect the

damping of the system, this vibration model can be expressed by the following equations:

$$\begin{aligned} M_1 \ddot{x}_1 &= -k_1 (x_1 - x_2) \\ M_2 \ddot{x}_2 &= k_1 (x_1 - x_2) - k_2 x_2 \end{aligned} \quad (1)$$

where, M_1 is the equivalent mass of the moving table plus the mass of the tooling that is placed on the top of moving table; M_2 is an equivalent mass of the base of the micro-positioning stage and two Klinger stages; k_1 is the equivalent stiffness of the flexure hinge mechanism in the vertical direction; k_2 is the equivalent stiffness of the Klinger stage system in the vertical direction. We assume that the two Klinger stages can be represented by a single mass-spring system. The modeling error due to this assumption is small because the stiffness of each Klinger stage is much higher than that of the micro-positioning stage (about fifteen times). The characteristic equation of Equation (1) is

$$\begin{vmatrix} -M_1 \omega^2 + k_1 & -k_1 \\ -k_1 & -M_2 \omega^2 + k_1 + k_2 \end{vmatrix} = 0 \quad (2)$$

solving for ω , the undamped natural frequency, gives:

$$\begin{aligned} \omega_{1,2}^2 &= \frac{k_1}{M_1} + \frac{k_1}{M_2} + \frac{k_2}{M_2} \\ &\pm \sqrt{\frac{k_1}{M_1} + \frac{k_1}{M_2} + \frac{k_2}{M_2} - 4 \frac{k_1}{M_1} \frac{k_2}{M_2}} \end{aligned} \quad (3)$$

Experimental measurement of the frequency response of the moving table showed that the positioning tower system had two resonant peaks at a frequency of 256 Hz and 432 Hz as seen in the Figure 3. To identify the source of these two vibration modes, we first used Equation (2) to solve M_1 and M_2 in terms of k_1 and k_2 . And then by measuring the stiffness of the micro-positioning stage and Klinger stages separately, we can calculate the equivalent masses of the micro-positioning stage and Klinger stages. Table 1 gives the measured stiffness, k_1 and k_2 , and the derived masses.

Table 1. Parameters of Micro-Positioning Tower System

	Stiffness (N/ μ m)		Mass (Kg)	
	k_1	k_2	M_1	M_2
Calculated	6.3	---	1.82	7.75
Measured and Derived Using Eq. (2)	5.3	49	1.73	7.88

From the table we can see that the values of derived masses using the above procedure are in close agreement with the calculated equivalent masses.¹ This, in turns, proves that the two natural frequency peaks shown in the Figure 3 are due to the vibration of the moving stage and the Klinger stages.

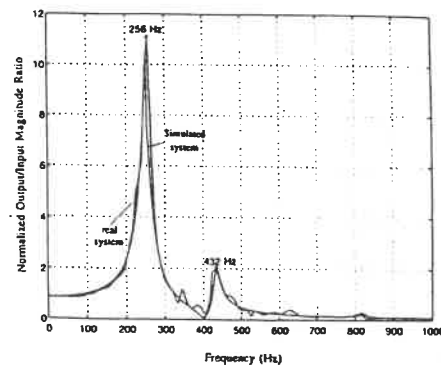


Figure 3. Simulated and Measured Open Loop Magnitude Response of the Micro-Positioning Tower System

Control System Simulation

The simulation analysis was based on the experimental data for the open loop magnitude and phase response of the combined system. First, we used the MATLAB package to obtain a transfer function that replicates the open loop frequency response of the combined

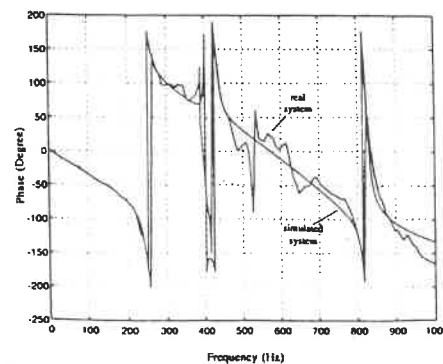


Figure 4. Simulated and Measured Open Loop Phase Response of the Micro-Positioning Tower System

micro-positioning stage system shown in *Figures 3 and 4*. The result showed that an 8th-order dynamic model gave better approximation to the real system than those with lower order dynamic models. Next, we designed a closed loop controller which combines a PID closed loop controller and 4th-order Chebyshev lowpass filter with 200 Hz cutoff frequency in series. The objective of using the lowpass filter is to suppress the open loop resonant peaks at 256 Hz and 432 Hz and, at the same time, maintain a flat frequency response as wide as possible.

Simulation of this closed loop controller on the computer model of the micro-positioning stage system showed that a flat frequency response in magnitude can be achieved over the frequency range of 0 to 80 Hz (see *Figure 5*).

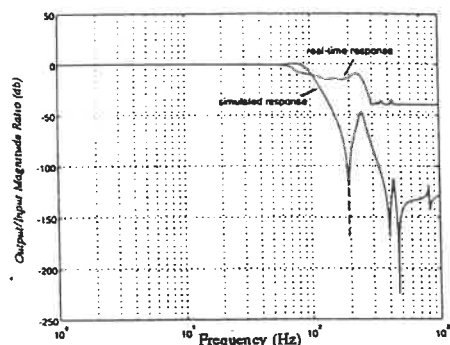


Figure 5. Frequency Response of the Simulated and Real-Time Closed Loop Control Systems of the Micropositioning Tower System

Real-Time Controller Design

Based on the computer simulation, a real-time control system was built to test the performance of the proposed controller. *Figure 6* shows a schematic of the control system setup. The corresponding block diagram for this control system is shown in *Figure 7*. In this setup, the digital control signal from the computer is sent

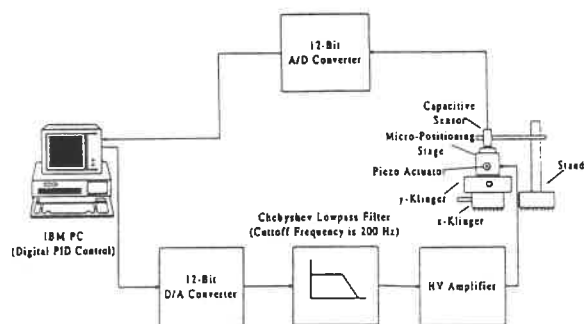


Figure 6. Control System Setup for the Micropositioning Tower System

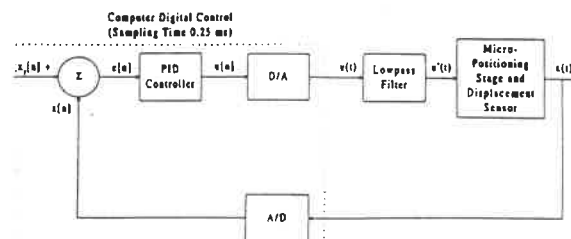


Figure 7. Block Diagram of the Control System for the Micropositioning Tower System

to the piezo-actuator inside the micro-positioning stage through first an analog 4th-order Chebyshev lowpass filter with cutoff frequency of 200 Hz and then through a high voltage amplifier with 0 to 1000 v output and X100 amplification. A noncontact capacitive-type sensor (Model 3910 from ADE Corporation, Newton, MA) with measurement range of $\pm 25 \mu\text{m}$, 5 kHz bandwidth and $2.5 \mu\text{m/v}$ resolution was used to detect the displacement of the moving table. The closed loop PID control parameters obtained from the simulation were taken as approximate values when controlling the system in real-time. Because of the modeling uncertainty, some adjustments were made in the control gains to achieve the desired frequency and time response performance.

To evaluate the closed loop control system performance, both frequency and time response tests were performed. A step input response test showed that a settling time of less than 0.04 second could be achieved for the designed closed-loop controller compared to a settling time of 0.12 second for the open loop system (see *Figures 8, 9*). In addition, the closed loop controller can effectively reject external disturbances. *Figures 10 and 11* show the open

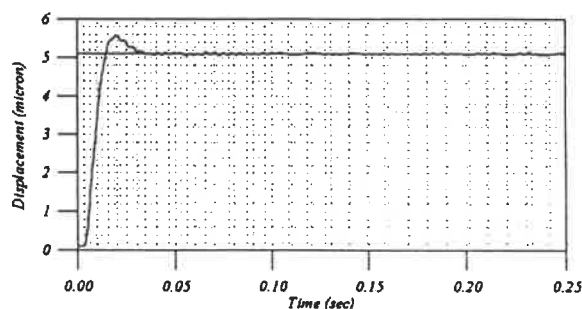


Figure 8. Closed Loop Step Response of the Combined Micropositioning Stage and the Two Klinger Stages

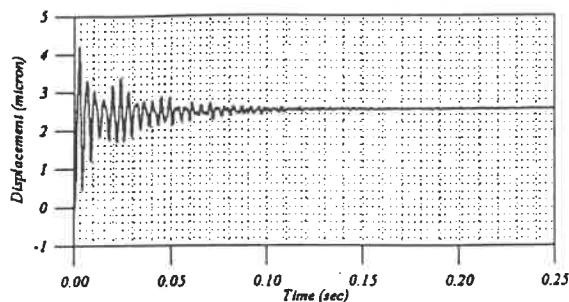


Figure 9. Open Loop Step Response of the Combined Micropositioning Stage and the Two Klinger Stages

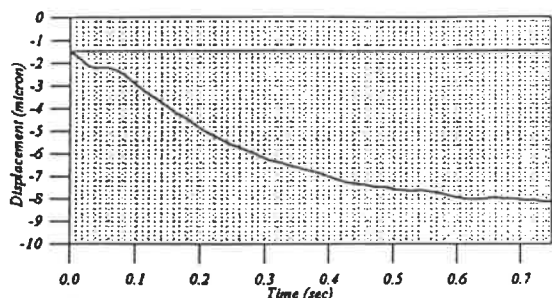


Figure 10. Open Loop Response of the Combined Micropositioning Stage and the Two Klinger Stages under an External Force Disturbance (The Force is Applied by Pushing Down on the Top of the Micropositioning Stage)

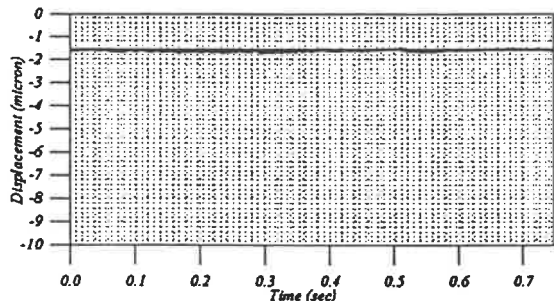


Figure 11. Closed Loop Response of the Combined Micropositioning Stage and the Two Klinger Stages under an External Disturbance (The Force is Applied by Pushing Down on the Top of the Micropositioning Stage)

loop and the closed loop system response, respectively, to an externally applied disturbance force. The force was applied by manually pushing on the top of the micro-positioning stage.

A magnitude frequency response test for the closed loop control system was performed by using white noise over the frequency range of 0 to 1000 Hz as the input signal, and the vertical displacement of the micro-positioning system as the output signal. The test showed that a flat frequency response in magnitude is achieved with a 3 db attenuation at 75 Hz (see Figure 5.)

The tracking performance of the

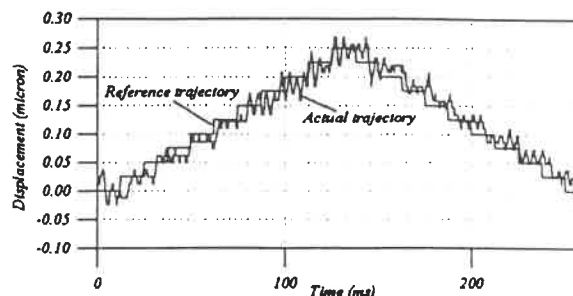


Figure 12. Closed Loop Control for Tracking a Step-Sinusoidal Input Signal

designed closed loop control system was tested by tracking a stepwise sinusoidal reference trajectory. Figure 12 shows the reference and actual trajectories. Noise from the lowpass filter limits the resolution of the system as seen in the Figure.

Conclusions

The design and implementation of a closed loop feedback control system for a micro-positioning tower made up of a z-motion micro-positioning stage and two Klinger stages was discussed in this paper. The control strategy is based on the use of a PID feedback controller with a 4th-order Chebyshev lowpass filter in series. The use of the Chebyshev lowpass filter suppresses the high frequency resonance modes of this multi-mode dynamic system while maintaining a flat frequency response in the low frequency range. This strategy significantly improves the dynamic behavior of the micro-positioning stage system. The strategy was tested in both simulation and real experimentation.

References

- [1] Yang, R., Jouaneh M., and Schweizer, R., "Design and characterization of a low-profile micropositioning stage", To appear in *Precision Engineering*, 1995.

Error Compensation of Multi-axis Manufacturing Center

Yiding Wang, Kee S. Moon, John W. Sutherland

Dept. of Mechanical Engineering--Engineering Mechanics

Michigan Technological University

Houghton, MI 49931. USA

Guoxiong Zhang, Yanfen Zang, Xiaohua Chu, Junwen Du

Dept. of Precision Instruments Engineering

Tianjin University

Tianjin, 300072. P.R.China

ABSTRACT

In this paper, the practice of error compensation of multi-axis manufacturing center is presented. The general geometrical error model of the machine is developed using homogeneous matrix method. The geometrical error compensation of a Cincinnati-Milacron manufacturing center was realized using this model and the compensation result is checked by measuring different diagonals in the machine working volume. For the thermal error compensation, a new thermal error model is put forward which use the "grey system" theory. This model has the ability to describe the long-term, internal characteristic of the non-linear system. The practice of the thermal error compensation of the spindle displacement is presented. A theory for the selection of the locations for the transducers is put forward. This theory not only can be used in the selection of optimal locations for transducers in thermal error compensation, but can also be used in the temperature control practice of the machine.

Keyword: error compensation, machine tool, accuracy, volumetric error, thermal deformation, grey system.

INTRODUCTION:

The combination of metrology with production control techniques is an important new characteristic of modern production engineering. With the development of computer and high precision sensors, more and more attentions have been paid to the improvement of manufacturing accuracy. Generally speaking, there are two factors which affect the manufacturing accuracy of the machine tools [Hocken,1980]: (1) quasi-static errors: which are the errors of the relative position between the tool and the workpiece that are slowly varying in time and are related to the structure of the machine tool itself. These errors may be divided into three general classes: those due to the geometry and kinematics of the machine, those due to static and slowly varying forces such as the dead weight of the machine components, over-constrained slides, workpiece weight, and the like, and those due to the thermally induced

errors in the machine tool structure. These errors cause the actual position to differ from the desired position which result in the dimensional errors of the workpiece. (2). Dynamic errors: these errors are very complicated. They mainly affect some micro characteristic of errors of the workpiece, such as surface finish, etc.

Only quasi-static errors of the machine are studied in this paper and compensated in the research.

There are generally two concepts for the accuracy enhancement of machine tools [Hocken 1980]: (1) Error avoidance: this is a means of error reduction in which the source of the error or its coupling mechanism is eliminated. It usually needs a large amount of investment on the hardware of the machine. (2). Error Compensation: this is a means of error reduction whereby the effect of the error is cancelled. Error compensation method is a very cost effective method.

GEOMETRIC ERROR MODEL OF THE MACHINE

Because we desired to make the error model of the machine as simple as possible, we used the rigid body assumption to obtain the error model of the machine [Zhang,1985]. In defining their errors, we always assume there are only six errors for each axis: one positioning error, two linearity errors, one pitch error, one roll error, and one yaw error. For a 3-axis machine tool we studied, there exists three squareness error between each axis. So the total error sources are 21.

In most cases, 3-axis machine tools can be classified into four groups. They are FXYZ, XFYZ, XYFZ, XYZF [Hocken,1980]. The letters before F denote available motion direction of the workpiece with respect to the fixed frame, and the letters after F denote the available motion directions of the tool with respect to the fixed frame. For example, in group FXYZ the workpiece is fixed and in group XYZF the tool is fixed.

The displacement of any point on a rigid body can be regarded as the combination of the linear displacement

of the point and the angular displacement of the point. In analyzing volumetric movement of rigid body, homogeneous matrix is a very efficient tool. The homogeneous matrix of a 3 dimensional volume is a 4x4 matrix [Donmez, 1986], that is:

$$Q = \begin{pmatrix} (C)_{3 \times 3} & (P) \\ 0 & 0 & 0 & 1 \end{pmatrix} \quad (1)$$

(C)_{3x3} denotes the angular error motion of x'y'z' coordinate system from the xyz coordinate system. (P) denotes the displacement of the reference point.

We regard X axis as the reference axis, XY plane as the reference plane. Thus, when the bridge moves a nominal distance x, the homogeneous matrix of this vector is:

$$Q(x) = \begin{pmatrix} 1 & -E_x(x) & E_y(x) & x \cdot \Delta(x) \\ E_x(x) & 1 & -E_z(x) & \Delta y(x) \\ -E_y(x) & E_z(x) & 1 & \Delta z(x) \\ 0 & 0 & 0 & 1 \end{pmatrix} \quad (2)$$

When the bridge moves to a nominal distance y, the homogenous matrix of the vector is:

$$Q(y) = \begin{pmatrix} 1 & -E_x(y) & E_y(y) & \Delta x(y) - \alpha_{xy} \cdot y \\ E_x(y) & 1 & -E_z(y) & y \cdot \Delta y(y) \\ -E_y(y) & E_z(y) & 1 & \Delta z(y) \\ 0 & 0 & 0 & 1 \end{pmatrix} \quad (3)$$

When the bridge moves to a nominal distance z, the homogeneous matrix of the vector is

$$Q(z) = \begin{pmatrix} 1 & -E_x(z) & E_y(z) & \Delta x(z) - \alpha_{xz} \cdot z \\ E_x(z) & 1 & -E_z(z) & \Delta y(z) - \alpha_{yz} \cdot z \\ -E_y(z) & E_z(z) & 1 & \Delta z(z) - z \\ 0 & 0 & 0 & 1 \end{pmatrix} \quad (4)$$

where E denotes the function of angular error, Δ denotes the function of positioning error, α denotes the squareness between axes.

We assume at the beginning of motion, all coordinate systems of the machine (fixed frame, workpiece, axes, tool) are aligned. In the actual manufacturing process, the vector of the tool tip must be coincided with the vector of the point to be machined on the workpiece

[Shultschik, 1977]. So if we define T(xt, yt, zt, 1) as the vector of the tip of the tool in the coordinate system of tool, P(xp, yp, zp, 1) as the vector of the machined point in the coordinate system of the workpiece. The following is true for a FXYZ type of machine:

$$P = Q(x)Q(y)Q(z)T \quad (5)$$

By substitute (2), (3), (4) to (5), the X,Y,Z component of the actual positioning error in the workpiece coordinate system can be obtained as:

$$\begin{aligned} & \Delta x = \Delta x(x) + \Delta x(y) + \Delta x(z) - y \cdot \alpha_{xy} - z \cdot \alpha_{xz} - y \cdot E_z(x) \\ & + z[E_y(x) + E_y(y)] - y[E_x(x) + E_x(y) + E_x(z)] \\ & + x[E_y(x) + E_y(y) + E_y(z)] + x_i \end{aligned} \quad (6)$$

$$\begin{aligned} & \Delta y = \Delta y(x) + \Delta y(y) + \Delta y(z) - z \cdot \alpha_{yz} - z[E_x(x) + E_x(y)] \\ & + x[E_x(x) + E_x(y) + E_x(z)] - z[E_x(x) + E_x(y) + E_x(z)] + y_i \end{aligned} \quad (7)$$

$$\begin{aligned} & \Delta z = \Delta z(x) + \Delta z(y) + \Delta z(z) - y \cdot E_z(x) - x[E_y(x) + E_y(y) \\ & + E_y(z)] - y[E_x(x) + E_x(y) + E_x(z)] + z_i \end{aligned} \quad (8)$$

For the other types of the machine, the positioning error formula can be obtained similarly, i.e. for XFYZ type machine, we have

$$Q(x)P = Q(y)Q(z)T \quad (9)$$

For XYFZ type machine, we have:

$$Q(x)Q(y)P = Q(z)T \quad (10)$$

For XYZF type machine, we have:

$$Q(x)Q(y)Q(z)P = T \quad (11)$$

GEOMETRIC ERROR COMPENSATION

We can realize the error compensation using the derived positioning error formula. First, we must obtain the geometric error terms of every axis as well as the squareness among the three axis. Then combine them using (6),(7),(8), thus we can obtain the positioning error of arbitrary point in the working volume by interpolation calculation. We then can make numerical correction based on the calculated errors by the software of the machine.

A few experiments for the calibration and error compensation on a CINCINNATI MILACRON 3-axis horizontal NC manufacturing center were conducted. By our classification method, it is an XFYZ type machine. The 21 errors of the machine can be obtained by HP-5528 interferometer. The thermal expansion of the carriages of the machine was measured by thermal transducers and combined into the geometric error terms. The compensation results can be checked by measuring different diagonals in the machine working volume. Our preliminary results showed a remarkable improvement

of the position accuracy of this machine.

THERMAL ERROR OF THE MACHINE.

The effects of thermal error loom so large in modern precision manufacturing. Many specialist world wide share a common view that the errors caused by the thermal effect are at least equal to the errors caused by the geometry imperfections if not larger [Bryan, 1990]. Thermal errors have the potential threat to the quality of the production and also cause some troubles in improving the efficiency of production. So the benefit of reducing the thermal errors is remarkable.

In theory, the thermal error is directly related to the temperature rise of the thermal source. The temperature field of the thermal source can be described by [Okushima, 1974]:

$$[C]\left\{\frac{dT}{dt}\right\}=[h][T]-[P] \quad (12)$$

[C]: thermal capacity matrix,
[h]: thermal impedance matrix
[p]: thermal load matrix,
[T]: temperature matrix

From thermal elastic theory, we have:

$$[F]=[G][T] \quad (13)$$

$$[F]=[K][U] \quad (14)$$

[F]: node force matrix,
[G]: transformation matrix of deformation-temperature.
[K]: rigidity matrix
[U]: thermal deformation matrix.

From (12), (13), (14) we can obtain the thermal deformation of the machine. So the thermal deformation can be obtained indirectly by measuring the temperature field changes of the machine.

THERMAL ERROR COMPENSATION MODEL OF THE MACHINE

The thermal deformation process of the NC machine is a non-linear, time varying, complicated system [Hocken, 1980]. The finite element method used by many scholars showed unsatisfactory result. One reason is that there are many assumptions in FEM which might not be true to the fact. The other reason is that it is very hard if not impossible to define the boundary conditions of this method. Besides, this method is time consuming and is hard to be used on line. Further more, thermal deformation has a sort of time-lag effect, that is, the current thermal deformation not only relates itself to the current state but also to the previous states (historical value). Hocken [Hocken, 1980] called this "memory of previous environment". FEM can not describe this

phenomena.

In our research, a new model which is called "grey system model" is used. We regard the system in which information is completely known as a "white system", and in which the information is completely unknown as "black system". For the system in which the information is not completely known, we call it "grey system" [Deng, 1987]. The grey system model is a kind of differential equation. It can describe the long term, internal developing trend of the system. The grey system model has the form of

$$\frac{d^n x_1^{(1)}}{dt^n} + a_1 \frac{d^{n-1} x_1^{(1)}}{dt^{n-1}} + \dots + a_{n-1} x_1^{(1)} = b_1 x_2^{(1)} + b_2 x_3^{(1)} + \dots + b_{h-1} x_h^{(1)} \quad (15)$$

The superscript (1) denotes that it is the first order accumulation of the variable. We regard the thermal deformation system of the machine tool as a grey system. In this system, the completely known information include: temperature change of the temperature monitoring points, thermal deformation of the deformation monitoring points, etc. Other information such as how the thermal flow is propagated in the machine etc. is not available. For the machine in our research, because of its symmetric structure, the most significant thermal deformation is the spindle displacement in the Y direction. We mainly discuss this displacement. So we assume the time series of the deformation in Y direction as X1, the time series of the temperature change of the temperature monitoring points as X2, X3 ... Xh, and obtain the grey system model of thermal deformation----temperature change of the temperature monitoring points. The grey system model was constructed using least square method.

We obtained the grey system model of the thermal deformation----temperature change of the temperature monitoring points using the data in the beginning 30 minutes after the machine started to work. Then predicted the errors of the future using the obtained model, and made numerical correction by the software of the machine. The result showed that the grey system model is accurate in describing the spindle thermal displacement of the machine. The error of the spindle displacement in the vertical direction can be compensated up to 85% from our first stage experiments.

THEORY FOR THE SELECTION OF TEMPERATURE MONITORING LOCATIONS IN THE THERMAL ERROR COMPENSATION

The earliest practices of selecting temperature monitoring locations were solely based on experiences and therefore were not efficient. With the development of transducer technology and data acquisition technology, people tend to use as many transducers as possible with the expectations of obtaining high

accuracy. But in practice, this is not beneficial to the result. One reason is that with more transducers used, the more the error source of the individual transducer will contribute to the final result. Besides, it is very inconvenient for the machine to operate properly with so many transducers placed on it. Also, it takes more time to process all the information provided by all the transducers, thus it is not good for on-line operation. So optimizing the selection of the locations for the transducers is beneficial to the simplification of the model and improvement of the reliability of the system. Last but not least, it is economical.

Here we briefly describe the theory for the selection of the locations for the transducers.

1. Pattern classification of the thermal behavior of the machine.

Different locations of the machine have different thermal behavior. First, we place as many as possible transducers on the machine to study the thermal behavior of the machine. We obtain the time series of the temperature change of every transducer and the time series of the thermal deformation of the point we want to study. We set up the AR model of the temperature change series as:

$$T(i-1)-a^T T(i)-\varepsilon(i) \quad (16)$$

$a:(a_1, a_2, \dots, a_n)$ are the coefficients of the AR model, $\varepsilon(i)$ is the white noise.

We set up the second order model and use the coefficient (a_1, a_2) as the characteristic parameter for pattern classification. The pattern classification was realized by using fuzzy ISODATA method. So the various transducers can be classified into a few groups and we can choose one transducer from every group as the characteristic transducer.

2. Optimizing the selection of the locations for transducers

Although the characteristic transducers are obtained, the function of every transducer in describing the thermal behavior of the machine is not the same. Our objective is to select the few points that will best describe the thermal behavior of the machine. In our study, we employ the so called "B type relevancy" in grey system theory. B type relevancy shows how close the relation is between the change of the process and the cause of the change of the process. We calculate the B type relevancy coefficient of the thermal deformation series to the temperature change series of every characteristic transducer. We select the few locations where the transducers have the biggest B type relevancy coefficient with the thermal deformation as the temperature monitoring points.

CONCLUSION

This paper describes an on-going research for the error compensation of the multi-axis manufacturing center. Error compensation method is a means to improve the accuracy of the machine without making many hardware changes of the machine. It is an economical method in improving the machine accuracy. Besides, error compensation method has a great impact on the design concept, that is the previous pursuit of higher accuracy can be directed toward the pursuit of stability and reliability [Zhang, 1985] so a good error budget can be achieved. Using error compensation methods, we can realize so called "precision machining without precision machinery" [Wu, 1989]. Future work should include: quick method for machine calibration, high quality software for error compensation, thermal error prediction and compensation of the machine in the real workshop conditions.

REFERENCE:

- Belfore, G. et al., 1987, Coordinate Measuring Machine and Machine Tools self calibration and error correction. *Annals of the CIRP*, Vol 36/1.
- Bryan, J., 1990, International status of thermal research. *Annals of the CIRP*. Vol 39/2.
- Bush, K., 1985, Calibration of coordinate measuring machines. *Precision Engineering*. No.2
- Deng, J., 1987, Grey control system. (in Chinese)
- Donmez, A. et al., 1986, A general methodology for machine tool accuracy enhancement by error compensation. *Precision Engineering*.
- Hocken, R. et al., 1977, Three dimensional metrology. *Annals of the CIRP*. Vol 25/2
- Hocken, R., 1980, Machine tool task force. Vol 5. UCRL-52960-5.
- Okushima, et al., 1974, An analysis of methods used in minimising thermal deformation of machine tools. *IMTDR*.
- Okushima, et al., 1975, Compensation of thermal displacement by coordinate system correction. *Annals of the CIRP*. Vol 24/1
- Shultschik, R., 1977, The components of volumetric accuracy. *Annals of the CIRP*. Vol 25/1
- Venugopal, R., 1986, Thermal effects on the accuracy of numerically controlled machine tools. *Annals of the CIRP*. Vol 35/1
- Wu, S. M., Ni, J. 1989 Precision Machining without precision machinery. *Annals of the CIRP*. Vol 38/1.
- Zhang, G., et al., 1985, Error compensation of coordinate measuring machines. *Annals of the CIRP*. Vol 35/1

A METHOD TO INTEGRATE NEURAL NETWORKS AND KINEMATICS BASED MODELS FOR MACHINE ERROR CHARACTERIZATION

H. M. Lee, J. Mou
Arizona State University
Tempe, AZ 85287-5906

1. Abstract

In this paper, an innovative approach is proposed to integrate both neural networks and kinematics based models for machine error characterization. A neural networks model is implemented first to map the nonlinear relationship among the positioning error, tool position, and temperature profile of the machine structure. The output of the neural networks model is the predicted coefficients for a kinematics based error model. The kinematics based model with the predicted coefficients is then used to compute the time-variant positioning error that relates to structural, kinematic, and thermally induced errors at any point within the workspace. In deriving the error model coefficients, there is no measured data is available for direct comparison with the neural networks output. Instead, the difference between the predicted errors from kinematics based model and the measured errors is used to train the neural networks. Therefore, a new training algorithm is derived for this integrated error modeling system by adopting the gradient-decent method. The issues of convergence, training rate, and number of hidden node are also discussed.

2. Introduction

Most multi-axis machines have axes that are designed to have unidirectional movement. Coordinated motion of these axes is required whenever it is necessary for the machine to move to a new position. Three types of errors occur during this movement: structural, kinematic, and thermally induced. Much of research has been conducted to characterize various machine errors for error estimation and compensation. The major assumption is that these errors are both repeatable and predictable. The goal is to improve the positioning accuracy of the machine to its repeatability limit.

Due to the complexity in structural configuration of various machines, it is not been possible to develop a single model for all machines nor to develop a signal model for each class of machines. By using rigid body kinematics, a family of models can be developed to relate positioning errors to structural and kinematic errors for given machines. The variation of error within a workspace is modeled as function of position. Once an error model is derived, the next task is the parameters estimation. Although neural networks and kinematics based models have been used for modeling machine errors, there is a lack of proper method to integrate both types of error modeling techniques for characterizing the machine errors that resulted from structural, kinematic, and thermally induced errors.

In this paper, an innovative approach is proposed to integrate both neural networks and kinematics based models for machine error characterization. The input and output of the neural networks are temperature profiles of the machine structure and the coefficients for the kinematics based model, respectively. The neural network model is used to map the nonlinear relationship among the positioning error, tool position, and temperature profile of a machine. With the derived coefficients and information of nominal tool positions, the time-variant positioning errors within the machine's workspace can be determined by using the kinematics based model. However, in deriving the error model coefficients, there is no measured data is available for direct comparison with the neural networks output. Instead, the difference between the predicted errors from kinematics based error model and the measured errors is used to train the neural networks. Therefore, a new training algorithm is needed for this integrated error modeling system. The criterion for neural networks training is to minimize the sum-of-squares-error of an energy function. For data-adaptive updating purpose, the gradient-decent direction method is adopted in deriving the training algorithm.

In following sections, the proposed integrated modeling approach is outlined and the training algorithm is derived. A case study is used to illustrate the general application of the integrated modeling approach. Finally, the issues relate to convergence, training rate, and number of hidden node are discussed.

3. Integrated Modeling Approach

In this section, an integrated modeling approach for machine error characterization is proposed. The major advantage of this modeling approach is that kinematics based model provides a systematic way to model, estimate, and compensate for the positioning error of a multi-axis machine. On the other hand, neural networks based model is particularly useful when a nonlinear mapping is required or the system of concern is too complicated to be modeled using classical linear approximation. In addition, neural networks are capable of deducing the underlying system properties from given data as well as to perform multiple input to multiple output pattern classification. All these

properties provide unique advantages for integrating both modeling techniques together for machine error characterization and correction.

As shown in Figure 1, two sub-models are interfaced together to form the integrated system. The first sub-model is a neural networks based model that provides the mapping between the temperature profile of the machine structure and the coefficients of the kinematics based sub-model. The inputs to the neural networks are thermocouple readings ($T_1, T_2, \dots, T_n$), where n is integer. The outputs are predicted coefficients for second sub-model. The second sub-model is derived by using kinematics and is used to compute positioning errors that relate to structural, kinematic, and thermally induced errors. The inputs to the second sub-model are nominal tool positions (x_1, x_2, x_3) and estimated parameters ($c_1, c_2, \dots, c_m$), where m is integer. The outputs are the predicted machine errors within the designated workspace. The difference between the predicted and measured errors is backpropagated to refine the weights of neural networks model. As the difference between the predicted and measured errors becomes negligible, the predicted errors can be used to generate proper error compensation signals for error correction.

4. Training Algorithm Derivation

Backpropagation is currently the most important and most widely used learning algorithm. The objective of its network training is to find the optimal weights to minimize the error between the training value and the actual response. The basic technique used in backpropagation is gradient decent method. In this section, a training algorithm is derived for the integrated system based on the formulas of backpropagation and gradient decent method. Suppose $E(\cdot)$ is a differentiable energy (or error) function that takes a set of network weights $W = \langle w_{ij} \rangle$ and produces a measure of the error, $E(W)$, for those weights. The error surface produced by $E(\cdot)$ gives the error for every set of weight values. Because $E(\cdot)$ is differentiable, its multidimensional derivative (or gradient) vector at any point within weight space can be computed as $\nabla E = \left\langle \frac{\partial E}{\partial w_{ij}} \right\rangle$. The gradient gives the direction in weight space that would

result in maximum increase of the error when an infinitesimally small weight change is made in that direction. Letting η be a small positive number, the revised weight vector W^* can be computed as $W^* = W + \eta \nabla E(W)$. Similarly, the vector in the opposite direction, $-\nabla E(W)$, would lead to minimizes error for a small step and the weight vector can be computed as $W^* = W - \eta \nabla E(W)$. This immediately suggests a learning algorithm: start at an arbitrary set of weights, W , and continue to evaluate $-\nabla E(W)$ while taking small steps in that direction until a stopping criterion has been reached.

The above learning algorithm is basically the backpropagation algorithm. A complete derivation can be found in *Digital Neural Networks* (Kung 1993). Here, the traditional backpropagation algorithm (BP1) is briefly summarized. A multilayer neural network model can be characterized by the following dynamic equations.

$$u_i(l) = \sum_{j=1}^{N_l-1} w_{ij}(l) a_j(l-1) \quad \& \quad a_i(l) = f(u_i(l)) \quad 1 \leq i \leq N_l; \quad 1 \leq l \leq L$$

where the input units are represented by $a_i(0)$, and the output units by $a_i(L)$. Also, L and N_l are the number of layers and the number of nodes in l -th layer, respectively. The activation function is very often a *sigmoid function* defined

as $f(u_i) = \frac{1}{1 + e^{-u_i}}$. The multilayer model is a function of inputs x and weights w and the energy function $E(x, w)$

is specifically chosen for an application. Then the weights vector w can be trained by minimizing the energy function along the gradient decent direction as $w_{ij}^*(l) = w_{ij}(l) - \eta \frac{\partial E}{\partial w_{ij}(l)}$, where η is a small positive number. The

error signal is defined as $\delta_i(l) = -\frac{\partial E}{\partial a_i(l)}$ and backpropagated through the neural networks during training process.

Algorithm BP1:

Step 1. Recursively obtain the error signal.

- Initial (Top) Layer: $\delta_i(L) = -\frac{\partial E}{\partial a_i(L)}$.
- Recursive Formula: The general recursive formula for the error signal $\delta_i(l)$ can be derived as $\delta_i(l) = -\frac{\partial E}{\partial a_i(l)} = \sum_{j=1}^{N_{l+1}} \delta_j(l+1) f'(u_j(l+1)) w_{ji}(l+1)$ in the sequence of $l = L-1, \dots, 1$.

Step 2. Recursively update the weights between the (l -th and ($l-1$)-th) layers (in the order of $l = L, L-1, \dots, 1$)

- $w_{ij}^*(l) = w_{ij}(l) + \eta \delta_i(l) f'(u_i(l)) a_j(l-1)$

Next, the training algorithm for the proposed integrated system is derived. Note that the input of the neural networks model is thermocouple readings. The coefficients $\{c_i\}$ are the output of the neural network model. The inputs of the kinematics model are $\{c_i\}$ and nominal tool positions. Let e_{x1} , e_{x2} , and e_{x3} be the computed positioning errors in $x_1(z)$, $x_2(y)$, and $x_3(x)$ axes respectively. Then e_{x1} , e_{x2} , and e_{x3} are functions of $\{c_i\}$. The notations e_{m1} , e_{m2} , and e_{m3} represent the measured positioning error. A training algorithm can be obtained by choosing a specific energy function and carefully manipulating the backpropagation algorithm BP1. However, in deriving the $\{c_i\}$, there is no measured data is available for direct comparison with the neural networks output. Instead, the difference between the predicted errors from kinematics based error model and the measured errors is used to train the neural networks. Therefore, the energy function is formulated as $E = 1/2[(e_{m1} - e_{x1})^2 + (e_{m2} - e_{x2})^2 + (e_{m3} - e_{x3})^2]$. The goal of the training is to minimize the above sum-of-squares-error criterion energy function. Since a special energy function is chosen, the error signals in algorithm BP1 can be reformulated as follows.

$$\begin{aligned} \delta_i(L) &\equiv -\frac{\partial E}{\partial a_i(L)} = \frac{\partial E}{\partial c_i} = \frac{\partial E}{\partial e_{x1}} \times \frac{\partial e_{x1}}{\partial c_i} + \frac{\partial E}{\partial e_{x2}} \times \frac{\partial e_{x2}}{\partial c_i} + \frac{\partial E}{\partial e_{x3}} \times \frac{\partial e_{x3}}{\partial c_i} \\ &= (e_{m1} - e_{x1}) \left(\frac{\partial e_{x1}}{\partial c_i} \right) + (e_{m2} - e_{x2}) \left(\frac{\partial e_{x2}}{\partial c_i} \right) + (e_{m3} - e_{x3}) \left(\frac{\partial e_{x3}}{\partial c_i} \right) \end{aligned}$$

The error positioning functions are assumed differentiable. Therefore, the gradient vectors of the error positioning functions e_{x1} , e_{x2} , and e_{x3} have important effect on error signals derivation. A reformulated version of propagation algorithm (BP2) is derived to train the proposed network.

Algorithm BP2:

Part 1. Using Kinematics based error model, error signals can be obtained and backpropagated.

- Output: $(e_{m1} - e_{x1})$, $(e_{m2} - e_{x2})$, and $(e_{m3} - e_{x3})$.
- Parameters: $(e_{m1} - e_{x1}) \left(\frac{\partial e_{x1}}{\partial c_i} \right) + (e_{m2} - e_{x2}) \left(\frac{\partial e_{x2}}{\partial c_i} \right) + (e_{m3} - e_{x3}) \left(\frac{\partial e_{x3}}{\partial c_i} \right)$ for each c_i .

Part 2. For neural network model training,

Step 1. Recursively obtain the error signal.

- Initial (Top) Layer: $\delta_i(L) =$ the error signal for parameter c_i .
- Recursive Formula: The general recursive formula for error signal $\delta_i(l)$ can be derived as

$$\delta_i(l) = \sum_{j=1}^{N_{l+1}} \delta_j(l+1) f'(u_j(l+1)) w_{ji}(l+1) \text{ in the sequence of } l = L-1, \dots, 1.$$

Step 2. Recursively update the weights between the (l -th and ($l-1$)-th) layers in the order of $l = L, L-1, \dots, 1$.

- $w_{ij}^*(l) = w_{ij}(l) + \eta \delta_i(l) f'(u_i(l)) a_j(l-1)$

The convergence behavior of algorithm BP2 is basically the same as the well-known backpropagation algorithm BP1 since BP2 is only a special case. The conventional wisdom is to keep η very small so as to assure eventual convergence. Convergence speed is only a secondary consideration. Also, this algorithm preserves the property of backpropagation learning. It allows the error signal of a lower layer to be computed as a linear combination of the error signal of the upper layer. In this manner, the error signals are backpropagated through all the layers from output layer of the proposed network.

5. Case Study

A set of experimental data is used to verify the feasibility of the proposed integrated modeling approach. The experiment was conducted at National Institute of Standards and Technology (NIST). A three-dimensional error model is derived to characterize the positioning error in $x_1(z)$, $x_2(y)$, $x_3(x)$ axes of a vertical machining center [Mou and Liu 95]. By using the derived algorithm, the neural networks model was trained with the given data set. This is done by backpropagating the derived error signals, based on the difference between measured and predicted positioning errors, to the neural networks model. For example, α_{11} is one of the coefficients of the kinematics based error model. Let's (x_1^*, x_2^*, x_3^*) be the nominal tool position; $(e_{x1}^*, e_{x2}^*, e_{x3}^*)$ be the predicted error with respect to a set of nominal tool

position and temperature profile; and $(e_{m1}^*, e_{m2}^*, e_{m3}^*)$ be the measured error with respect to (x_1^*, x_2^*, x_3^*) and as the training data for the proposed network.

$$\text{If } \frac{\partial e_{x1}}{\partial \alpha_{11}} = 0; \frac{\partial e_{x2}}{\partial \alpha_{11}} = \sum_{i=2}^4 a_i x_1 + 1/2x_1^2 + 2x_1x_3; \text{ and } \frac{\partial e_{x3}}{\partial \alpha_{11}} = 2x_1x_2 - \sum_{i=2}^4 b_i x_1.$$

$$\text{Then, } \Delta \alpha_{11}^* = \eta [(e_{m2}^* - e_{x2}^*) (\sum_{i=2}^4 a_i x_1^* + 1/2(x_1^*)^2 + 2x_1^* x_3^*) + (e_{m3}^* - e_{x3}^*) (2x_1^* x_2^* - \sum_{i=2}^4 b_i x_1^*)].$$

The training set consists of 60 input-output pairs. Various numbers of hidden nodes and learning rates have been used to train the neural networks. After a number of trails, we found that the networks with 30 hidden nodes, learning rate 0.01, momentum 0.01, noise 0.5, and 50000 iterations can converge to 0.0013 inch. This result is not good enough to our requirement. The reason might be that kinematics based model is used for error estimation and we may facing the problem of solving some singular matrices for feasible coefficients. We are currently exploring a method to effectively select proper data set in avoiding the possible singularity problem.

6. Reference

Kung, S. Y., 1993, Digital Neural Networks, Prentice-Hall Inc., Englewood Cliffs, New Jersey.

Mou, J. and C. R. Liu, 1995, "A Methodology for Machine Tool Error Correction Based on Reference Parts," *International Journal of Computer Integrated Manufacturing*, Vol. 8, No. 1, pp. 64-77.

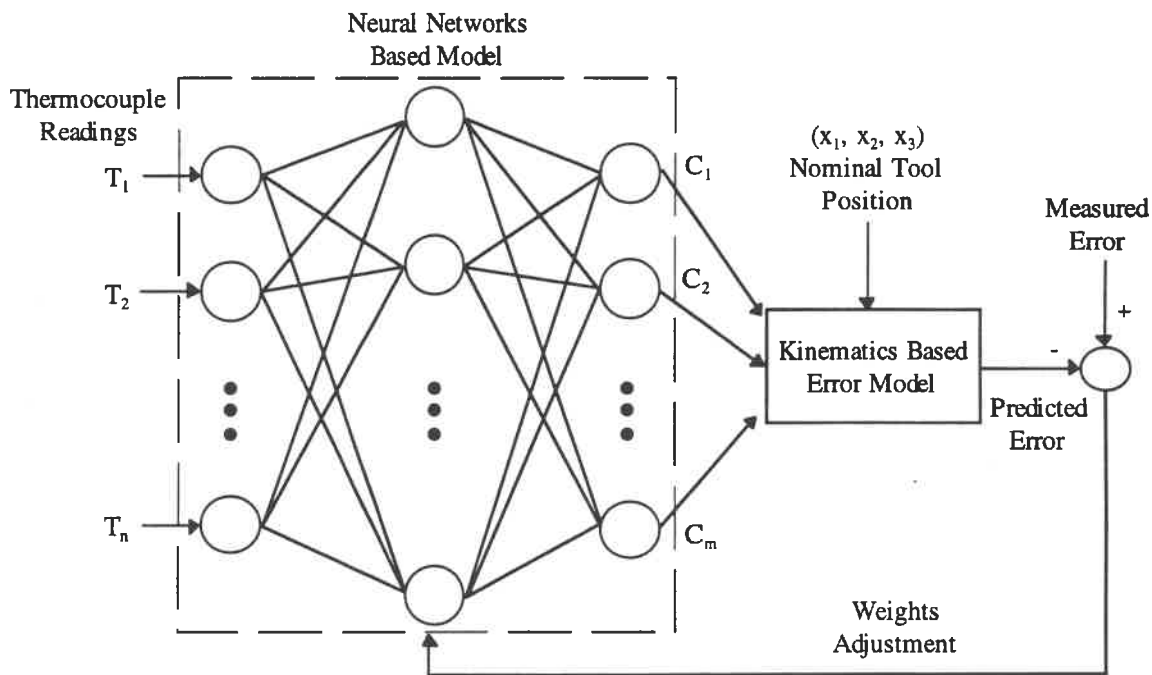


Figure 1. Integrated Error Modeling System for Machine Error Characterization

Digital Compensation of Sensors With Frequency Output

Tian Guiyun

**(Measurement & Control Engineering Department
Sichuan Union University, Chengdu, Sichuan 610065 /P. R. China)**

1. Introduction

Sensor systems for mechanronic infrastructure developed very quickly. The circuits trending in the realisation of integrated sensors and converters are increasing demand for easy communication with a microprocessor. In many cases the conversion of a sensed physical quantity into a digital number is indispensable because compensation, calibration and linearization are accomplished with a microprocessor. For this purpose the measured physical quantity is converted to pulse with modulated (PWM) or frequency modulated (FM) signal, which can be regarded as an intermediate step between analogue and digital signals. There are many advantages in using frequency conversions of the measured physical variance.

It is important for us to design electric oscillates for the mechanatronic sensor of displacement with frequency output. The oscillates change electronic paramters(R-resister, L-inductor, C-capactor) into frequency quantities. The research key is the stability of the oscillate. Using the oscillates, we research inductive sensors and eddy current sensors success. The mechanatronic sensor of displacement with frequency output is very small and useful. They are used for contact or non-contact measures.

The sensors with frequency output can transfer analogue information into time domain and can be digitised by a microcotroller directly. It also eliminates the need of expensive AD converter. Therefore this kind of sensors is useful and is developing very quickly, especially at integrated and smart sensors. FD converter, linearisation and compensation are very important for the sensors with frequency output.

2. Design and experimental detail

2.1 Oscillators

In order to get the frequency output, it is important for us to design the LC oscillators with high stability for sensors. For high resolution about inductive sensors, the oscillator frequency must be increased and the coil induction or the capacitance reduced. In the inductive probe design, we change the technology of the amplitude-modulated (AM) probes by omitting the magnet steel (or magnetiser) around the coils. This not only simplifies the probe structure and technological process, but also overcomes the eddy loss

and magnetic hysteresis loss effectively . It gives the transducer high stability and low zero drift.

In order to work on differential operation, we use the differential solenoid inductive probe. Two coils have different oscillatory frequencies, and they change oppositely, in respect to the change of the displacement. It is important to overcome the frequency implicative motion. New coils and cylindrical ferrite beads were adapted, the coil tuning elements are two cylindrical ferrite beads 2.8mm in diameter by 4mm long, they are interval by 2mm long. the beads were commented to the end of a copper stem, another end is attached the measuring plant. the beads are located at the centre of the solenoid coaxially.

The system performance directly depends on the stability of two LC oscillators. On the basis of the theoretical analysis³, We researched the oscillating conditions and stability, pronounced the fresh evidence and adapted many methods to stabilise the frequencies.

2.2 Measuring frequency and digital compensation

Generally, Sensors with frequency output have low dynamic frequency response. We design the high speed and high precision measuring frequency circuits called multi circle syno counting. For example, we can use 8253 IC which has three 16-bit counters. Counter 0 ,1 are used for divider to measured frequency(f_x). The output of counter 0 ,1 acts as gate controller to control counter 2 for measuring high frequency (f_ϕ) which comes from crystal oscillators. If the number of divide frequency is N_x , the counter 2 can count f_ϕ within N_x circles of measured frequency (f_x). Therefor, the result N_ϕ of counter 2 is:

$$N_\phi = f_\phi \cdot N_x / f_x \quad (1)$$

$f_\phi / f_x \gg 1$, if the resolution of sensors is constant, the method of multi circle syno counting can improve the frequency response of sensors with frequency output greatly.

With Eq (1), If computers are used for control measurement and calculation directly, it waste measuring time and computer resoure. We adapt EEPROM for register and " run reading " the counter results (Fig. 1).

$$N_{\phi_{n+1}} - N_{\phi_i} = f_\phi \cdot (N_{x_{n+1}} - N_{x_i}) / f_x \quad (2)$$

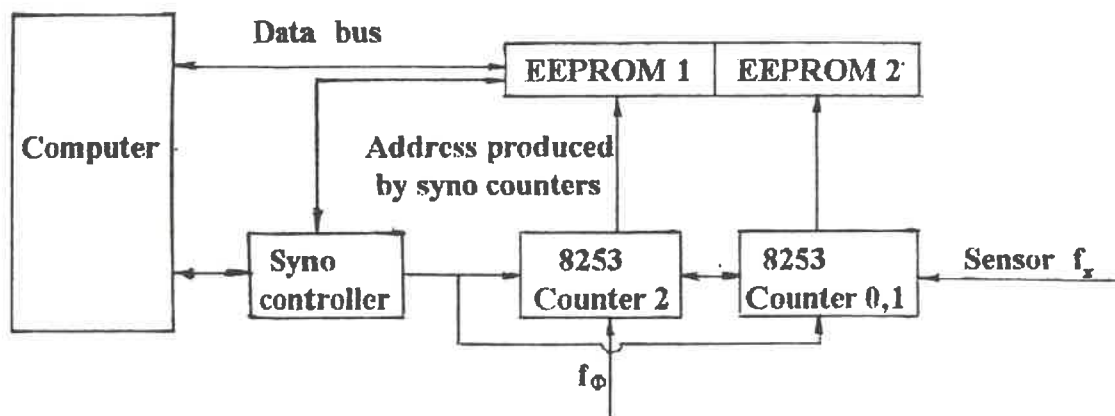


Fig. 1 Implement of multi circle syno counter and compensation circuit

Measurement arm beam: $[measBeam] = ([PBS_{meas}] [\theta_{retro}] [PBS_{meas}]) e^{-i(OPD)}$ (4)

where the optical path difference (OPD) is given as $OPD = \left(\frac{2\pi}{\lambda}\right) 2d$. (5)

The laser wavelength (λ) is the wavelength of the measurement beam, which is frequency shifted from the reference beam for a heterodyne interferometer and d is the measurement cube corner's displacement with respect to the reference cube corner. An exact treatment would require propagating the leakage term with a slightly different wavelength (due to the dual-frequency nature of the laser source), but the resulting errors are small and will be ignored here.

Finally, the interferometer matrix is simply the sum of the reference and measurement beam Jones matrices, where one or both beams can be attenuated due to misalignments, etc.:

$$[Interferometer] = \kappa_r [refBeam] + \kappa_m [measBeam]. \quad (6)$$

Here, κ_r and κ_m represent attenuation in the reference and measurement beams, respectively.

Laser source

For the heterodyne interferometer, the matrix representation should include the effects of non-orthogonality of states, ellipticity, and angle with respect to the given coordinate system axis. Once again, using Jones vector form:

$$[LaserSource_{AC}] = [\theta_{laser}] \left[[\theta_{nonorthog}] A_1 \begin{bmatrix} 1 & e^{-i(\omega_1 t)} \\ \alpha_1 & e^{-i(\omega_1 t - (\frac{\pi}{2})}) \end{bmatrix} + A_2 \begin{bmatrix} \alpha_2 & e^{-i(\omega_2 t - (\frac{\pi}{2})}) \\ 1 & e^{-i(\omega_2 t)} \end{bmatrix} \right] \quad (7)$$

where α is the coefficient of ellipticity, A represents the amplitude coefficient of each frequency component, $\theta_{nonorthog}$ is the angle of nonorthogonality and θ_{laser} is roll orientation of the laser head's axis with respect to the coordinate system defined by the beamsplitter. The angle of nonorthogonality rotates only one frequency component of the two frequency laser source, while the laser head angle rotates the entire coordinate system of the laser radiations with respect to the interferometer.

Detection

The electric field vector incident on the detector is then

$$[E_i] = [Interferometer] [LaserSource_{AC}]. \quad (8)$$

The detector consists of a polarizer (at an angle) and an intensity detector. In the heterodyne case, the final electric field vector is given as

$$[E] = [-\theta][P][\theta][E_i]. \quad (9)$$

Normally, in an AC interferometer, the polarizer angle is set at 45° for heterodyne detection of beat signals. This angle ensures a mixing of the polarization coded reference and measurement beams at the detector. This final, complex electric field vector is transformed into an intensity scalar by pre-multiplying the Jones vector by the complex conjugate of the matrix transpose.

Symbolic algebra programming

Using the *Mathematica*³ programming language, the algebraic manipulation of terms was automated so that the final outcome of the simulation was an exact, symbolic expression. Actual simulation values such as wavelength and displacement are only substituted locally within plotting statements.

For the heterodyne interferometer, the intensity signal resulting from Jones matrix multiplication was first expanded as a complex series using Euler's identity and then simplified to its lowest form. A *Mathematica* substitution rule is invoked to replace any term containing a form of $\omega_1 t - \omega_2 t$ with

$(\omega_1 - \omega_2)t$. The resultant was then expanded using trigonometric identities to resolve the $\sin^2$ terms. The resulting many-termed polynomial is then tested term-by-term to recover any term containing $(\omega_1 - \omega_2)t$, which is then appended to a new polynomial containing only the beat signal terms that are functions of $\Delta\omega t$. The final polynomial is a sum of sinusoids which are all functions of $\Delta\omega t$.

To recover phase independently from signal amplitude variations, a quadrature mixer scheme was implemented in the model as illustrated in Fig. 1.

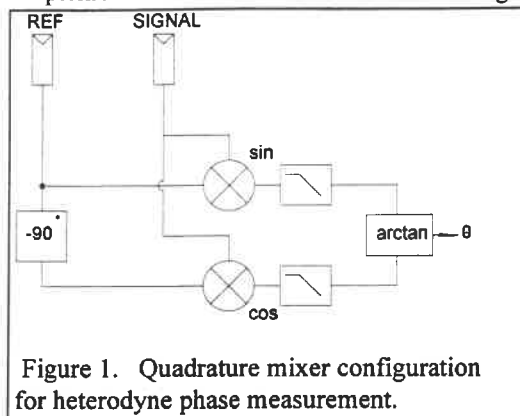


Figure 1. Quadrature mixer configuration for heterodyne phase measurement.

The beat signal was simply multiplied by sin and cos reference signals at the frequency $\Delta\omega t$, the result then expanded using trigonometric identities to eliminate quadratic terms, and all the non-baseband terms are discarded using a similar testing algorithm to the one used for beat signal recovery. In this case, only the terms 'mixed' down in frequency to baseband are kept while the harmonic terms are discarded. This procedure is done for both quadrature signals. The final phase information is recovered by simply computing the arctangent of the resulting quadrature signal. It was somewhat reassuring, at this point, that when all the error terms were turned off, the final outcome was an arctangent term dependent upon displacement and the computed error terms summed to exactly zero.

Application

To test the interferometer model, three polarization mixing observation or compensation schemes were simulated using Jones Matrix models^{4,5,6}. In all cases, the model produced results similar to those previously published. To simulate one experimental situation⁶ more completely, we modified the model slightly to observe results in the time domain. Polarization mixing can be observed as an amplitude modulation of the beat signal given a fixed mirror velocity. In the computer model, the terms containing displacement are simply replaced with velocity multiplied by time using a *Mathematica* rewrite rule. The given output modulation terms may be directly observed, or a spectrum analyzer can be simulated by computing FFTs to view the results in the frequency domain, as in Fig. 2. The vertical scale is in decibels to show relative amplitudes of the heterodyne signal components. 4096 sampled points were evaluated and a Blackman window applied to the data to improve signal to noise ratio. Note that the amplitude of the Doppler shifted beat signal remains constant and the first order error terms appear at the nominal beat frequency. By observing the simulated spectra of various interferometer error sources, it is noted that individual errors leave distinct signatures.

Polarization Mixing Error Spectra

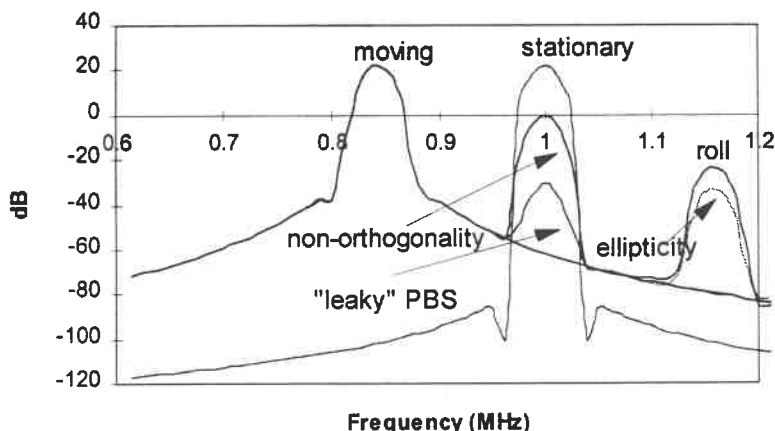
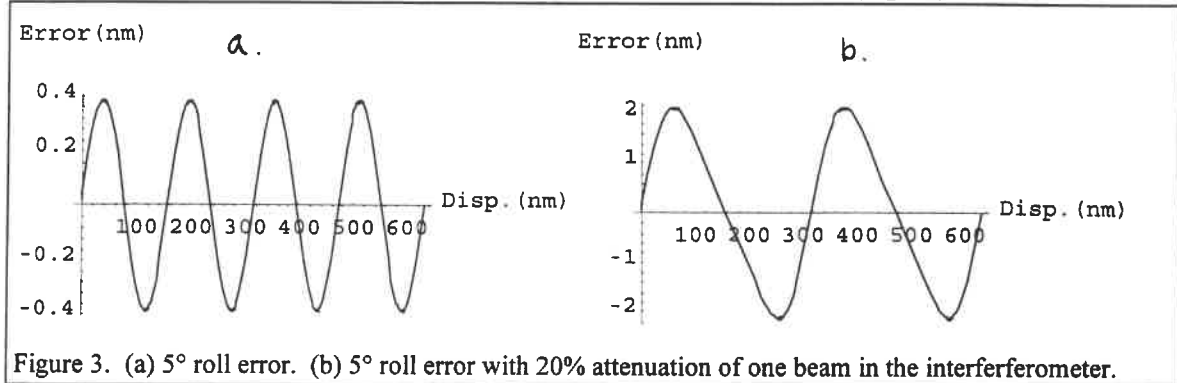


Figure 2. Error spectra of 4 different types of heterodyne interferometer error.

In Figure 2, the beat frequency is 1 MHz, the mirror velocity is $50 \text{ mm}\cdot\text{s}^{-1}$, the leaky polarizing beamsplitter is defined to be 10,000:1 in the *S* state and 500:1 in the *P* state. The angular errors of roll and non-orthogonality are both given as 4 degrees and the ellipticity is 4%. These values were chosen as representative of a typical system before undergoing any optimization. Note that the logarithmic scale of the vertical axis exaggerates the importance of the other error terms. Only non orthogonality is capable of producing errors of more than several nanometers in most instances.

Using the model, it was discovered that two error conditions can combine to produce interferometer deviations that are greater than merely the sum of their parts. For example, a simple laser head/interferometer roll axis error of 5° produces 0.8 nm of second order error (Fig. 3a). Combining this 5° roll with a 20% reduction of the electric field in one of the interferometer arms increases the error to more than 4 nm and changes the nature of the error from second to first order (Fig. 3b).



It is interesting to note that using a laser head producing unbalanced intensities in the polarization states (such as those produced by an acousto-optic modulator inside the laser head) has no effect on the error susceptibility to roll. Only attenuations in the arms of the interferometer produce this compounding of errors. Such beam attenuations can be easily caused by incorrect interferometer alignment, however.

DC or homodyne interferometers may also be modeled using similar techniques, as well as Fabry-Perot cavities (and combinations thereof). The simple ease with which a simulation may be performed, about a 30 s task, encourages experimentation. More error terms can be incorporated in the model until theory is in better agreement with experimental practice.

¹ R.C. Jones, *J.O.S.A.*, 31, 488, (1941).

² R. Azzam and N. Bashara, *Ellipsometry and Polarized Light*, (North-Holland, New York, 1977), p.66.

³ The identification of commercial products is given only for the sake of completely describing our experimental procedures. In no instance does such identification imply recommendation by the National Institute of Standards and Technology, nor does it imply that the particular equipment or software identified is necessarily the best available for the described purpose.

⁴ A.E. Rosenbluth and N. Bobroff, *Prec Eng*, 12, 7, (1990).

⁵ W. Hou and G. Wilkening, *Prec Eng*, 14, 91, (1992).

⁶ S. Patterson and J. Beckwith, *Proceedings of the 8th International Precision Engineering Seminar*, (Elsevier Science, Inc., 1995), p.101.

Dynamic Tool Path Determination System Based on the Experimentally Obtained Stability Lobe Diagram Using Multi-Axis Force

Mamoru MITSUISHI, Takashi OHTA, Hideyuki OKABE and Takaaki NAGAO

Department of Engineering Synthesis, Faculty of Engineering

The University of Tokyo

Hongo 7-3-1, Bunkyo-ku, Tokyo 113, Japan

1. Introduction

A manufacturing system which integrates a machining center with a multi-axis force sensor and real-time controller, CAD system and data-base was developed to realize high precision and high efficiency autonomously. In this paper, an end-mill machining in a machining center is discussed. To realize highly efficient machining while maintaining a stable cutting state, the following two functions were implemented: (1) a real-time adaptive control function based on multi-axis force sensor information and (2) a predictive control function using a physical model of cutting to determine the machining conditions and tool path. It is, however, difficult to predict the phenomena accurately because the actual cutting phenomena are more complicated than the physical models in the data-base. Therefore, learning capability was added in the system to reflect the actual state of the system by feeding back the result of adaptive control to the data-base.

2. Related work

Many researchers have developed physical models of end-milling [1, 2, 3, 4, 5]. In particular, Takata et al. have introduced the idea of a machining scenario to change the machining conditions when the geometry of a workpiece varies during the tool path [1]. In the system, optimization of the machining conditions is accomplished before machining. The problem of monitoring tool wear or breakage has been approached through the use of acoustic emission sensing [6, 7], video image analysis, accelerometers, pulse generators or heat flux measurement. Neural networks have been tried recently for signal processing of sensor signals [6, 7, 9]. Concerning chatter vibration, we have a good review by Sato[10]. Smith and Tlustý [11] have presented a method for machining state detection and the modification of the machining conditions. However, it has the problem that it takes

several seconds to judge the machining state using FFT. The "stability lobe diagrams," which plot the boundaries separating stable and unstable machining under constant machining conditions, are discussed in, for example, reference [12]. Generally, the diagram is obtained from a static machining condition model. In this paper, in contrast, the stability lobe diagram is experimentally obtained. Furthermore, this paper discusses a system which can deal with varying machining conditions. This is important, since conditions change frequently in real parts, due to changes in the geometrical model of the mechanical part. Such adaptation is possible using our real-time machining state detection method because the method is fast enough to use force data during a single tool rotation.

3. System construction

Fig.1 shows an overview of the system described in this paper. The system consists of a CAD system, a data-base part and a real-time controller. One of the features of the system is that these processes are not static, but dynamic, communicating with each other during the cutting sequence. More concretely, the contents of the data-base are dynamically revised by being sent the actual cutting conditions and cutting state from the real-time controller. Furthermore, the three-dimensional modeller in the CAD system can be accessed dynamically during cutting. Tool path and cutting conditions are regenerated depending on the revision of data-base and are sent to the real-time controller. In the real-time controller, the real-time machining state judgment method [13] is used to realize adaptive control.

4. Data-base

The data-base consists of an "experimentally obtained stability lobe diagram," a "machining efficiency evaluation task" and a "machining condition determination task." The stability lobe diagram introduced by Merritt[12] and Tlustý is

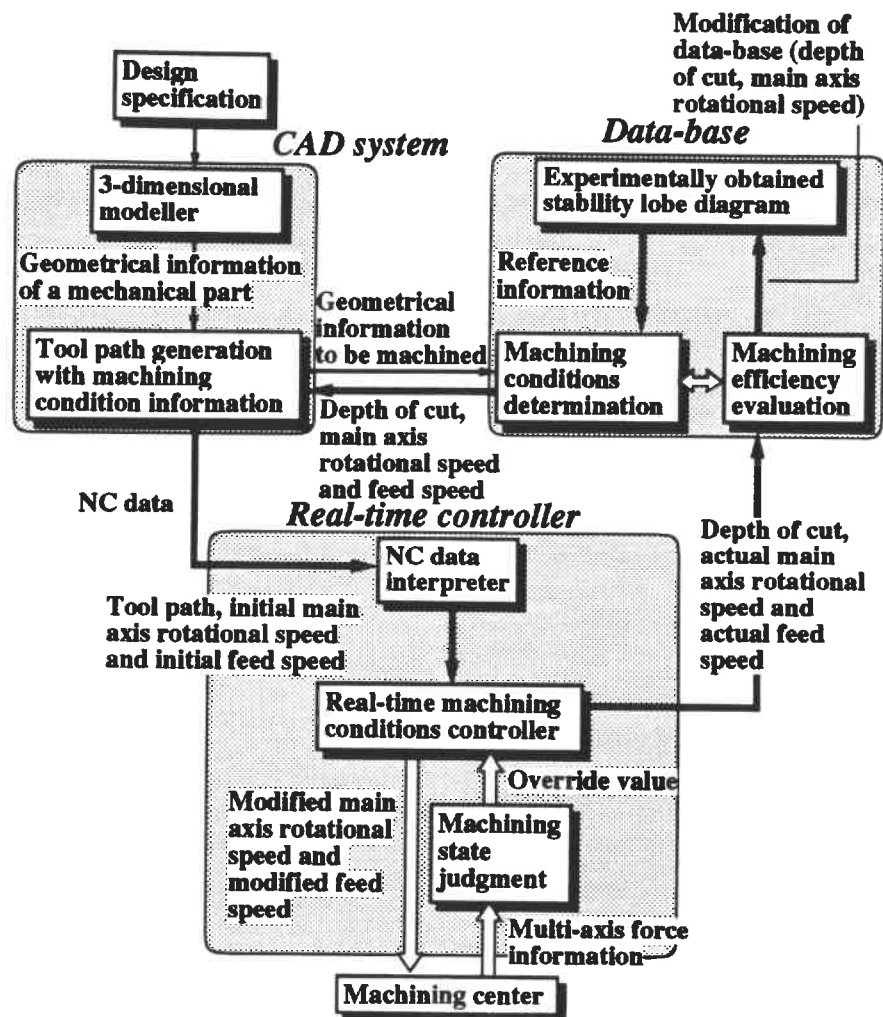


Fig.1 System configuration

experimentally obtained in the system described in this paper[14]. Therefore, it is called an "experimentally obtained stability lobe diagram." To obtain the stability lobe diagram, the dynamics of the structural loop which consists of tool, workpiece and machine tool has to be analyzed. In general, the dynamics of the structure is experimentally obtained by methods such as (i) excitation by tapping, (ii) the impulse response method, and (iii) oscillation tests. To apply method (i), it is difficult to realize the same conditions such as excitation point, time duration and direction. Method (ii) is still being researched. Consequently, method (iii) is most commonly used, although it is very time-consuming. For practical use it is important that the stability lobe diagram be obtained. Therefore, a cutting operation is actually executed by a machine tool and the stability lobe diagram is automatically obtained by judging the machining state in real-time using force information.

The machining efficiency evaluation task and the machining condition determination task use the 3-dimensional modeller in the CAD system to determine efficient machining conditions and generate an appropriate tool path. Furthermore, the machining efficiency evaluation task modifies the stability lobe diagram as it varies during machining.

5. Real-time control

In the real-time controller, adaptive control is executed based on the real-time machining state judgment method[13]. The "NC data interpreter task" acts as an interface between the CAD system and the real-time controller. The NC data interpreter task accepts any arbitrary NC data set which is described in the APT format. The "real-time machining condition controller" and the "real-time machining state judgment task" perform adaptive control by adjusting the override values for main axis rotational speed and feed speed based on the previously determined

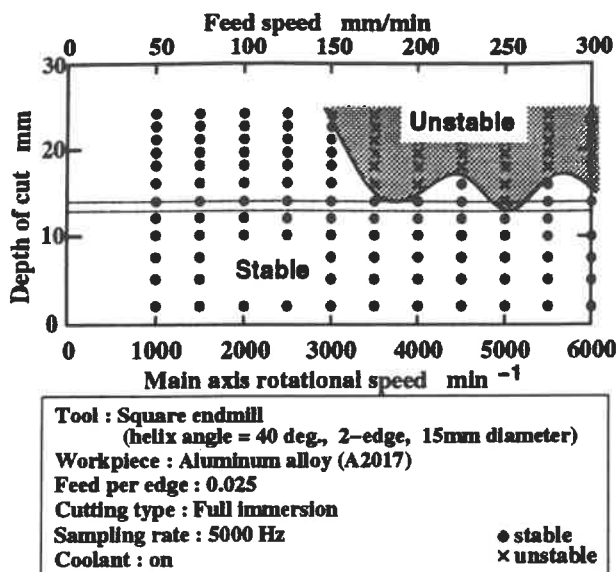


Fig.2 Stability lobe diagram

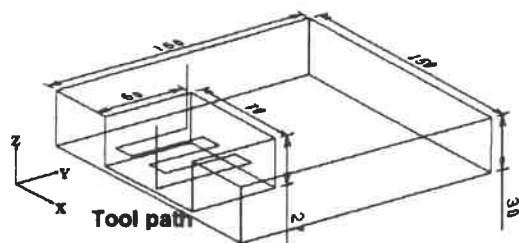


Fig.3 Shape of a workpiece

machining conditions. Furthermore, the relationship between the machining conditions and the machining state is fed back to the data-base.

6. CAD system

In the CAD system, geometrical information for the component mechanical parts of a product is input to the 3-D modeller and the abstracted machining feature information is sent to the machining condition determination task in the data-base. The stability of the endmill machining system depends on the main axis rotational speed and the depth of cut, according to the stability lobe diagram theory[12]. Therefore, the machining conditions determination task returns the most efficient set of main axis rotational speed and depth of cut for the given machining geometrical features. The tool path generation task generates a 3-dimensional tool path referring to the received depth of cut and geometrical machining feature information.

The most efficient set of main axis rotational speed and depth of cut obtained from the machining condition determination task changes when the stability lobe diagram in the data-base

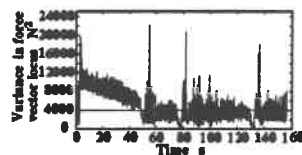


Fig.4 Variance of distance in force vector locus

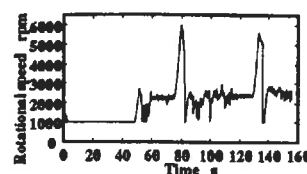


Fig.5 Main axis rotational speed (1)

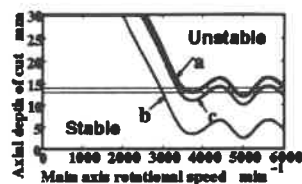


Fig.6 Variance of stability lobe diagram

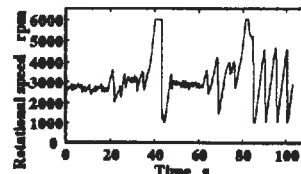


Fig.7 Main axis rotational speed (2)

is modified. Therefore, in the system described in this paper, the tool path generation task is invoked periodically during machining and the tool path and cutting conditions are quasi-dynamically modified to maintain the most efficient cutting. Typically, the tool path is regenerated after each cutting pass.

7. Experimental results

Cutting experiments were performed to evaluate the system. The following rules were applied in the adaptive real-time controller: (i) The main axis rotational speed and the feed speed are increased to increase efficiency while the cutting state is stable. (ii) The main axis rotational speed and the feed speed are decreased to avoid chatter, when it occurs. Rule (ii) takes priority over rule (i). The main axis rotational speed and the feed speed are proportionally controlled in the proposed system to maintain the feed per edge constant and maintain a constant surface roughness of the cut. The geometrical model of a workpiece is shown in Fig.3. In the experiment, the cutting conditions were the same as those shown in Fig.2.

The machining condition determination task estimated depth of cut of 14 mm and main axis rotational speed of 6,000 rpm as appropriate for the first cutting pass, towards the total depth of cut of 27 mm. Fig.4 shows the variance of distance in the force vector locus [13] when the experiment was performed under the conditions mentioned above.

Between 0 and 50 seconds, chatter vibration occurred because the tool was not sufficiently worn to provide frictional damping and preventive control was executed. The main axis rotational speed could be significantly increased at

about 80 seconds and 130 seconds because light cutting conditions were encountered when the tool was moving in the X-direction in Fig.3 and the width of cut was locally decreased. From Fig.5 it is also clear that between 108 seconds and 125 seconds the cutting state was stable and cutting conditions remained almost constant. From this result, the system recognized that 2,940 rpm is the stable-unstable boundary for cutting with a depth of cut 14 mm, which was obtained after 108 seconds. This result was used to modify the data-base, by the modification method described in the next section.

8. Dynamic change of the data-base

Fig.6 shows the variance of the experimentally obtained stability lobe diagrams based on the experimental results mentioned above. "Curve a" shows the initial estimated stability lobe diagram determined from the data-base and derived from other similar experiments. For the current experiment, the stability lobe diagram was modified in "curve b" to fit the actual conditions for a depth of cut of 14 mm and a main axis rotational speed of 2,940 rpm, the limiting speed of the stable-unstable boundary. In the implemented system, it is also assumed that stable-unstable boundary moves up and down depending on the tool wear process. In the experiment, the machining condition determination task was invoked again for the remaining 13 mm depth which remained for the overall, desired cut depth of 27 mm. The system determined that the machining condition should be 2,996 rpm for the main-axis rotational speed and 13 mm for the depth of cut. These machining conditions were used as initial conditions for the 2nd cut pass and adaptive control was performed again. Fig.7 shows the variance of the main axis rotational speed. Between 0 and 85 seconds, the initial cutting conditions were maintained almost constant even though the adaptive control was applied. The result shows that the previously determined cutting conditions were still appropriate. However, after 85 seconds, the system gradually evolved to a new stable-unstable boundary at 4,640 rpm in the main axis rotational speed for a 13 mm depth of cut, as shown "curve c" in Fig.6.

9. Conclusions

A learning function can be added to the system by feeding back the results of the adaptive controller, which maintains a stable cutting state based on experimental information obtained by multi-axis force sensing. Efficient machining conditions, including the detailed tool path, were dynamically realized by regenerating the tool path periodically from information in

a data-base which describes the relationship between the cutting conditions and cutting state. Experimental results showed the effectiveness of the system. Future work will develop techniques to find the globally optimum cutting conditions in the case where there are several discrete stable solutions for the predefined depth of cut. Work will also be conducted to develop a common data-base, useful for many machine tools.

References

- [1] Takata, S., et al., "Generation of a Machining Scenario and Its Application to Intelligent Machining Operations," *Annals of the CIRP*, 42/1, pp.531-534, 1993.
- [2] Tlustý, J., et al., "New NC Routines for Quality in Milling," *Annals of the CIRP*, 39/1, pp.517-521, 1990.
- [3] Minis, L., et al., "Analysis of Linear and Non-linear Chatter in Milling," *Annals of the CIRP*, 39/1, pp.459-462, 1990.
- [4] Armarego, E.J.A. et al., "Computerized End-Milling Force Predictions with Cutting Models Allowing for Eccentricity and Cutter Deflections," *Annals of the CIRP*, 40/1, pp.25-29, 1991.
- [5] Altintas, Y. et al., "End Milling Force Algorithms for CAD Systems," *Annals of the CIRP*, 40/1, pp.31-34, 1990.
- [6] Moriwaki, T. et al., "Application of Neural Network to AE Signal Processing for Automatic Detection of Cutting Tool Life," *Journal of JSPE*, vol.57, no.7, pp.1259-1264, 1991.
- [7] Dornfeld, D.A., "Neural Network Sensor Fusion for Tool Condition Monitoring," *Annals of the CIRP*, 39/1, pp.101-105, 1990.
- [8] Lundholm, T., et al., "Advanced process monitoring — A major step towards adaptive control," *Robotics and Computer-Integrated Manufacturing*, vol.4, no.3, pp.413-421, 1988.
- [9] Moriwaki, T. et al., "Recognition of Cutting State Based on Neural Network Sensor Fusion," *Journal of JSPE*, vol.59, no.5, pp.779-784, 1993.
- [10] Sato, H., "Vibration problem of a machine tool," *Journal of JSME*, vol.97, no.911, pp.848-853, 1994.
- [11] Smith, S. et al., "Stabilizing Chatter by Automatic Spindle Speed Regulation," *Annals of the CIRP*, 41/1, pp.433-436, 1992.
- [12] Merritt, H.E., "Theory of Self-Excited Machine Tool Chatter," *ASME J. Eng. for Industry*, pp.447-454, Nov. 1965.
- [13] Mitsuishi, M., et al., "Real-time Machining State Detection Using Multi-axis Force Sensing," *Annals of the CIRP*, 41/1, pp.505-508, 1992.
- [14] Mitsuishi, M., et al., "Automatic Acquisition of Machining Conditions: Towards a Self-Evolving Machining Center," *Proc. of the 9th Annual Meeting of the American Society for Precision Eng.*, pp.338-341, Cincinnati, U.S.A., 1994.

**ACCURACY IMPROVEMENT OF MACHINE TOOL
BY WORKPIECE-REFERRED CONTROL
(4TH REPORT)**

- Sensor Position for Plain Turning -

Takanori Yazawa, Yutaka Uda, Tsuguo Kohno
Tokyo Metropolitan Institute of Technology
6-6 Asahigaoka, Hino, Tokyo 191 Japan

1. Introduction

In ultraprecision machining technology, manufacture of more precise and larger elements is expected. In the case of turning, the machined form accuracy is determined by the so-called copy principle. General accuracy improvement of machine tool has considerable difficulties from an economical point of view. The author proposes a new system WORFAC (Workpiece Referred Form Accuracy Control) which is a way of controlling the depth of micro-cutting via in-process measurement between tool and workpiece[1]. In the previous report[2], it was found out that simulation of WORFAC is effective for estimating the formed accuracy. In this study, the positioning of sensor is investigated both simulation on a computer and experiment of plain turning.

2. Principle of the research

The WORFAC is based on a concept that form accuracy is determined by the relative displacement between the workpiece and the cutting tool. Hence this method is a way of controlling the depth of micro-cutting via in-process measurement of the distance between the tool and the workpiece. The experimental system for plain turning is illustrated in Fig.1. A micro-tool servo (MTS) and a noncontact optical surface sensor (HIPOSS) are attached rigidly on the same post, and the workpiece is placed between the MTS and HIPOSS. The MTS consists of parallel leaf springs, a piezoelectric actuator, and strain gauges.

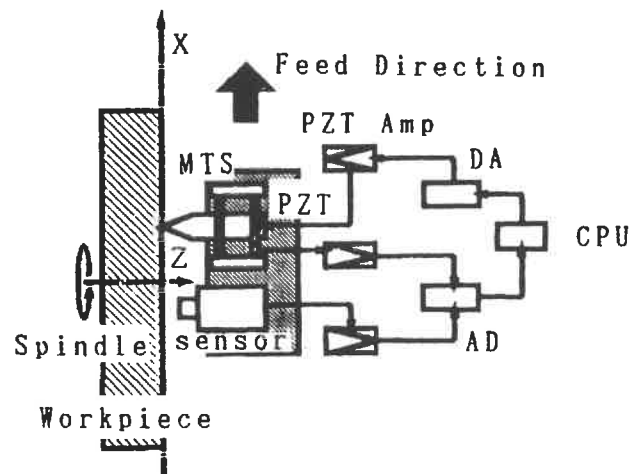


Figure 1 Block Diagram of
WORFAC for Plain Turning

3.How to simulate WORFAC

For simulation of the WORFAC on a computer, the mechanism of plain turning is modeled and shown in Fig.2, and assumed as the cutting is regarded as ring like not herical. The X-axis is defined as the feed direction and the cutting origin is the center of rotation. Y and Z-axis are defined as shown in Fig.2. x is the position of cutting point and the tool post motion of the depth of cut direction will be appeared as an error $T_0(x)$. In the case of servo-off turning, the displacements of the cutting tool, the sensor, as well as the machined workpiece figure are all the same as to $T_0(x)$. In the case of servo-on turning, the motion of tool makes figure change from $T_0(x)$ to $T(x)$ but the motion of sensor is still $T_0(x)$. Here, x_g and y_g are the offsets between sensing point and cutting point. r is the distance from the origin to the sensor and written by the equation (1).

$$r = \sqrt{(x - x_g)^2 + y_g^2} \quad (1)$$

Because of the assumption, we are able to be supersede sensing point $P(x - x_g, y_g)$ by $P(r, 0)$, and the sensing amount $S(r)$ can be written by Eq.(2).

$$S(r) = T_0(x) - T(r) + C \quad (2)$$

Where C is the offset volume of the sensor in the cutting direction and is set so that $S(r)$ becomes zero at the starting point x_0 of servo-on turning. Thus $T(x)$ is expressed by the Eq.(3).

$$\begin{aligned} T(x) &= T_0(x) - S(r) \\ &= T(r) - C \end{aligned} \quad (3)$$

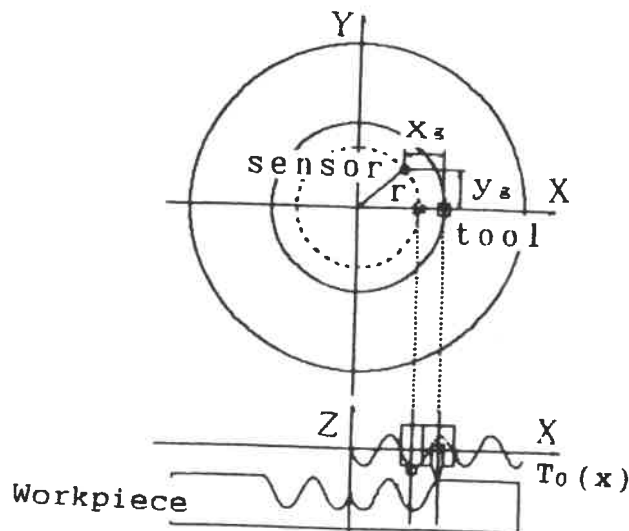


Figure 2 Simulation Model of equipment

4. Investigation of sensor positioning

The Eq.(3) shows that new surface is produced by sum of old reference surface and $-C$. Since the new surface is regarded as the next reference surface, the next controlled surface is made by sum of previous surface form and $-C$. So, C is heaped up and it may cause to make new error. For making $C=0$, the control should be started as soon as a sensor and a tool reach on a concentric circle simultaneously, i.e., $x_0 = r$. To prove the above condition, we make use of Eq.(2) in the case of control start point, and substitute $x_0 = r$, obtain,

$$\begin{aligned} S(r) &= S(x_0) \\ &= T_0(x_0) - T_0(x_0) + C \\ &= 0 \\ \therefore C &= 0 \end{aligned} \tag{4}$$

Hence, on simulations and experiments, hereafter, the sensor is started when a sensor and a tool reached on a concentric circle. Figure 3 is the results of simulations. As shown Fig.3, when $y_g = 0$, the form accuracy is the wrongest. Growing y_g bigger, the error becomes smaller at the area of $y_g < x_g$ and in the case of $y_g = x_g$, the best result is obtained.

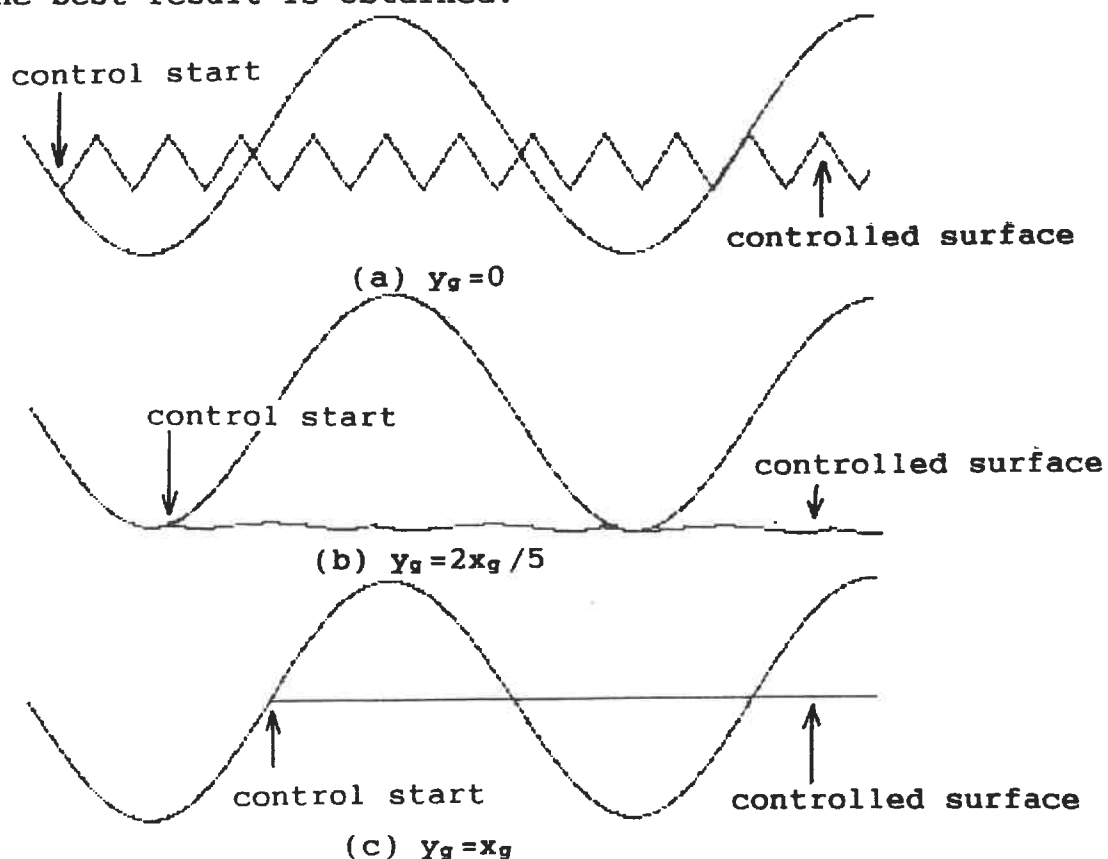


Figure 3 Results of Simulation

The results of plain cutting are shown in Fig.4. The cutting conditions are as follows: spindle speed 1000rpm, $d=10\mu\text{m}$, $f=20\mu\text{m/rev}$. The form accuracies are measured by a form tester. Their results are similar results as indicated by simulation.

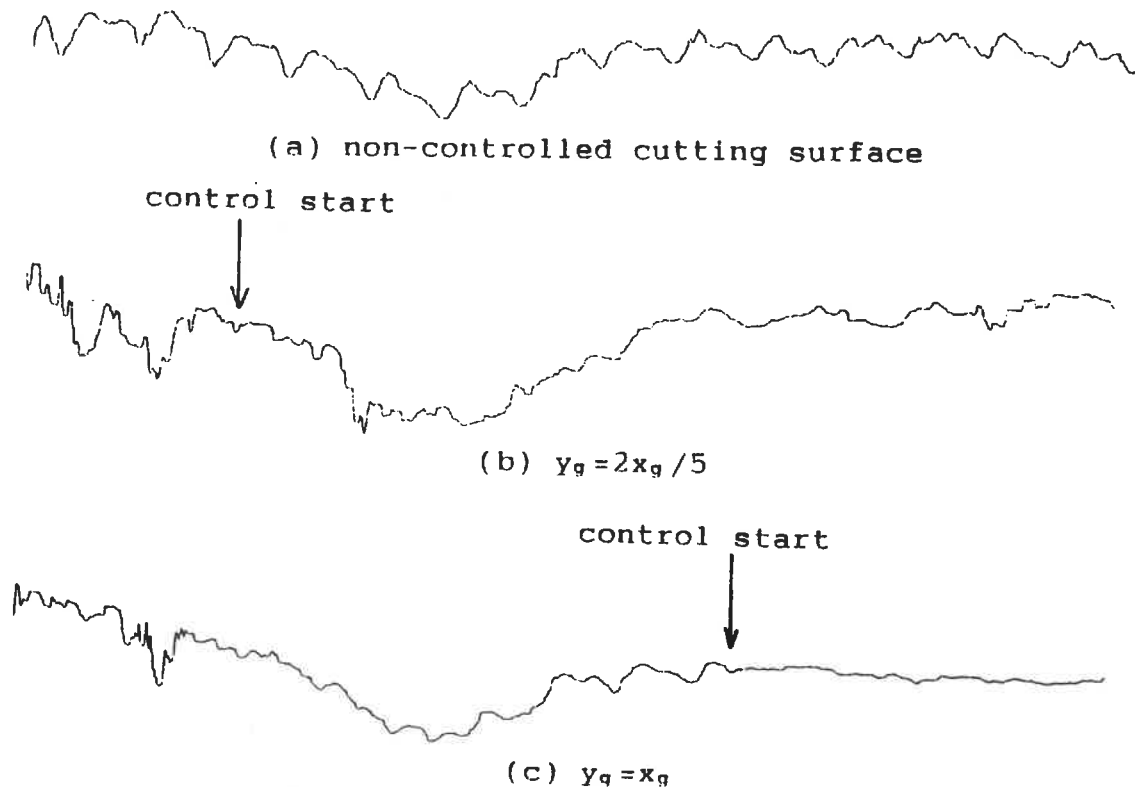


Figure 4 Results of Plain Turning

5. Conclusion

Workpiece-Referred Form Accuracy Control (WORFAC) is applied to a plain turning, and a simulation for the case is carried out on a computer. The results of simulation and experiment are compared and the sensor positioning is investigated.

(1) Growing the offset on Y-axis or y_g bigger up to $y_g = x_g$, the error becomes smaller.

(2) In the case of $y_g = x_g$, the sensor positioning is the best.

(3) $y_g > x_g$, the start point becomes late.

Reference

- [1] T. Kohno, et al. Precision Engineering, P.9, Vol.11, No.1, 1989
- [2] T. Kohno, et al. Proceedings of ASPE Annual Meeting '92, P.376
- [3] T. Yazawa, et al. Advancement of Intelligent Production, '94, P.233

DYNAMIC HYBRID POSITION/FORCE CONTROL OF DEBURRING ROBOT MANIPULATORS

R.Venkata Ramana
Research Scholar

T.Nagarajan
Professor

Y.G.Srinivasa
Associate Professor

Department of Mechanical Engineering,
Indian Institute of Technology, MADRAS - 600 036. INDIA

1. Introduction

When the burrs are predictable uniform ones, there will be no demanding on the deburring tool for force/position control. In other words, high strength steels or super alloys etc are difficult to machine and produce burrs of variable characteristics. In such situations, the deburring tool has to follow the following problems: 1. Position Control of the manipulator along directions of natural force constraints. 2. Force control of the Manipulator along direction of natural position constraints. This Hybrid Control has been established by extensive experimental settings [1],[2],[3] and others. This paper reviews some of the relevant work, and presents the simulation procedure for a two-arm manipulator.

Key Words Hybrid control, Deburring, robot.

2. Design of Hybrid Control System

The model for the hybrid force/position control [4] is an enhanced version of the model developed by Raibert et al. The linear adaptive force control law is given by:

$$F(t) = T_r(t) + c(t) + K_f(t) \int_0^t G(t) dt + K_p(t) G(t) - K_v(t) \dot{Z}(t) \quad (1)$$

where, $G(t) = T_r(t) - T$, $\dot{G}(t) = -\dot{T}(t)$, $\ddot{G}(t) = -\ddot{T}(t)$

where $F(t)$ is the $m \times 1$ force vector applied to the end-effector in the force sub-space $\{Z\}$, $T_r(t)$ is the required contact force on the burr used as a feed-forward term, $c(t)$ is an auxiliary signal associated with the feed-forward term, Z is the $m \times 1$ vector of end-effector position, $K_v(t) = K_D(t)K$ is the $m \times m$ velocity feedback gain vector, $K_v(t)Z(t)$ is the velocity damping, $\{K_p(t), K_i(t), K_D(t)\}$ are adaptive gains of the PID controller.

The terms of the force control model like $c(t)$, $K_i(t)$, $K_p(t)$ and $K_v(t)$ adapted to the system are given by:

$$c(t) = c(0) + \delta_1 \int_0^t p(t) dt + \delta_2 p(t) \quad (2)$$

$$K_i(t) = K_i(0) + \alpha_1 \int_0^t p(t) G^*(t) dt + \alpha_2 p(t) G^*(t) \quad (3)$$

$$K_p(t) = K_p(0) + \beta_1 \int_0^t p(t) G(t) dt + \beta_2 p(t) G(t) \quad (4)$$

$$K_v(t) = K_v(0) - \gamma_1 \int_0^t p(t) Z(t) dt - \gamma_2 p(t) Z(t)$$

$$\text{where, } p(t) = S_f G^*(t) + S_p G(t) - S_v Z(t) \quad (5)$$

$$G^*(t) = \int_0^t G(t) dt$$

The position Control System is given by:

$$F(t) = f(t) + K_p(t) G_p(t) + C(t) U(t) + B(t) \dot{U}(t) + A(t) \ddot{U}(t) \quad (6)$$

where $U(t)$ is the reference position trajectory vector, $G_p(t) = U(t) - Y(t)$ is the $n \times 1$ position tracking-error vector, $f(t)$ is an auxiliary signal. The adaptation laws required to adapt the auxiliary signal and controller gains to the task are given by:

$$f(t) = f(0) + \delta_1 \int_0^t t dt + \delta_2 u(t) \quad (7)$$

$$K_p(t) = K_p(0) + \gamma_1 \int_0^t u(t) G_p(t) dt + \gamma_2 u(t) G_p(t) \quad (8)$$

$$K_v(t) = K_v(0) + \eta_1 \int_0^t u(t) G_v(t) dt + \eta_2 u(t) G_v(t) \quad (9)$$

$$c(t) = c(0) + \mu_1 \int_0^t u(t) U'(t) dt + \mu_2 u(t) U'(t) \quad (10)$$

$$B(t) = B(0) + \gamma_1 \int_0^t u(t) U'(t) dt + \gamma_2 u(t) U'(t) \quad (11)$$

$$A(t) = A(0) + \gamma_1 \int_0^t u(t) U'(t) dt + \lambda_2 u(t) U'(t) \quad (12)$$

3. Simulation Procedure for a two arm manipulator

Consider the end-effector is moving in the x-direction, and let F_x be the control force applied to the tool. The following values are assumed for force control : $\alpha_1 = 150$, $\beta_1 = 25$, $\gamma_1 = 5000$, $K_I(0) = 1500$, $K_p(0) = 150$, $K_v(0) = 6000$ and $p(t) = g_x^*(t) + 0.5 g_x(t)$ and all other coefficients are zeros. The following values are assumed for position control in y direction : $\gamma_1 = 0.7$, $\eta_1 = 0.7$, $\mu_1 = 0.1$ and all other coefficients are zeros.

The simulation will be set up by mapping the cartesian forces acting at the hand into equivalent joint torques as:

$$\tau(t) = J'(\theta) \begin{bmatrix} F_x \\ F_y \end{bmatrix} \quad (13)$$

The performance of adaptive force/position control scheme is done with the following equation:

$$\bar{\tau}(t) = D \ddot{\theta} + h + J' \tau(t) \quad (14)$$

Finally the following are to be specified: a. The constant force set-point T_r along x-direction and b. The equation of the position trajectory $Y_r(t)$. Both of these are to be implemented simultaneously by the adaptive controller.

4. Conclusions

When the contact stiffness is very high in robotic deburring, it is necessary to implement a hybrid control for both force and position of the tool w.r.t. the work-piece. This paper reviewed some of previous relevant research work. A hybrid control has been presented and the simulation procedure has been given for a two arm manipulator.

REFERENCES

1. P.G.Backes, et al, Real-Time Cartesian Coordinate Hybrid Control of a PUMA 560 Manipulator, Proc. IEEE In. Conf. on Robotics and Automation, pp. 608-613,1985.
2. John J.Craig et al, Hybrid Position/Force Control of Manipulators, ASME J.Dyn.Sys.&Meas. Control, vol.102. 1981.
3. A.A.Goldenberg et al, Constrained Motion Task Control of Robotic Manipulators, Mech.Mach. Theory Vol.29,No.1, 1994.
4. O.Khatib, J. Burdick, "Motion and Force Control of Robot Manipulators", Proc. IEEE Int. Conf. on Robotics and Automation, San Francisco, CA, 1986.

Investigation of Acoustic Emission for Use as a Wheel-to-Workpiece Proximity Sensor in Fixed-Abrasive Grinding[#]

J. S. Taylor*, M. A. Piscotty*, D. A. Dornfeld[‡], K. L. Blaedel*, L. F. Weaver*

^{*}Lawrence Livermore National Laboratory

[‡]University of California at Berkeley

Summary

This paper reports on the feasibility of using Acoustic Emission (AE) for sensing the proximity of a grinding wheel to a glass workpiece, both prior to contact and in the early stages of contact. Our measured AE signals indicate that we can track the position of the grinding wheel as it approaches the workpiece through the turbulent coolant layer and then as contact initiates with a workpiece during spherical generation. Our data for the initial contact region is dominated by cyclical bursts of AE that appear to correspond to tool spindle motion errors. Our principal goal is to minimize the time required to 'find the part' without damaging the surface of a brittle workmaterial, i.e. during the transition from a fast approach to the much slower final in-feed required for the grinding operation. Our results also suggest that AE is useful as a gauging signal in determining the position of the grinding wheel with respect to the machine tool.

Introduction & Motivation

The Center for Optics Manufacturing (COM) and the American Precision Optics Manufacturers Association (APOMA) are undertaking a program to modernize the US optics industry, with a focus on introducing CNC fixed-abrasive grinding technology to companies that have relied on traditional multi-part blocks and loose-abrasive grinding¹. One of our goals in collaborating with COM is to better harness advanced sensors and control strategies for improving process economics and workpiece quality for this new generation of precision CNC optics grinders.

For CNC spherical generation, grinding cycle time is dominated by the accumulation of in-feed times. For a single operation, the in-feed can be divided into five steps: 1) fast approach, 2) approach leading to contact, 3) grinding, 4) dwell (zero), and 5) fast retraction. The grinding operation usually accounts for the longest duration of in-feed (about 1 minute). However, for medium and fine grinding, the time required for the approach to contact (2) may be about the same duration as for grinding, depending on the methodology. If the rate of in-feed during this phase of 'finding the part' is low, then productivity is reduced by what has been dubbed 'air grinding'.²

A reliable proximity sensor, coupled with feedrate over-ride control for the machine tool, could greatly reduce the time to 'find the part'. One approach is to use a force sensor and detect proximity by sensing the increase in force due to tool-to-workpiece contact.² Alternatively, acoustic emission may be used in the same mode of contact detection and for monitoring the 'spark-in' process.³ In fact, Table 1 lists a number of commercial sources of AE systems for detecting tool-to-workpiece contact.

In this paper, we consider the use of AE sensors for determining the proximity of the grinding wheel to the workpiece *prior to the initial contact with the workpiece*. We feel this is an appropriate strategy for the fine and medium grinding of glass optical surfaces and other brittle materials, where excessive subsurface damage might be created by initial high-speed contact.

Approach

We conduct our grinding development on a T-base diamond turning lathe that has been converted to a spherical generator. Glass workpieces are cemented to holders held in a collet in the

[#] This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract No. W-7405-ENG-48.

10-inch air bearing part spindle mounted on the z-axis. Bound-diamond cup wheels are mounted in collet-taper adapters for the high-speed air bearing tool spindle located on a rotary table on the x-axis. The machine tool has laser interferometer position feedback with 1 μ inch resolution. The nominal operating conditions for many of our tests are given in Table 2.

During testing, we cement an AE sensor to the back of a workpiece, with the leads extending back through the part holder, to a pre-amplifier mounted inside a cavity in the spindle. The basic set-up of our equipment is shown in Fig. 1. The signal and power lines from the pre-amplifier extend through the back of the spindle to a set of slip rings. Either within the pre-amplifier or as an auxiliary stage, we typically use a band-pass filter to eliminate aliasing (e.g. 1.2 MHz) and to minimize low frequency variations. The signals are typically viewed on an oscilloscope, and then are sent to a personal computer (Apple Macintosh) using a data acquisition package (National Instruments Labview). Once collected within the computer, we perform various mathematical calculations, including RMS power, filtering, and power spectral density (Matlab from the Mathworks). As indicated in the figure, the amplified signal is also sent to an envelope detector, which rectifies and filters the signal, producing a relatively-smooth positive signal that is proportional to AE signal power.⁴

Our testing procedure generally involves measuring AE signals with respect to a defined position of 'contact' between the wheel and the workpiece. Our approach to defining the contact position is to perform a grinding operation followed by a long-term dwell period (> 1 minute). At this point, we assume that material removal is essentially complete, and that most of the elastic deflection of the machine structure has relaxed (via material removal). We define this condition of long-term dwell as the zero-separation condition. Clearly, this is an imperfect definition, because of the uncertainties

Table 1. Vendors of AE Proximity Systems.

Euchner-USA Inc. Hibernia, NJ TEL: (201) 586-2600	Gap Eliminator
Montronix Ann Arbor, MI TEL: (313) 677-7890	TS100
Physical Acoustics Corp. Princeton, NJ TEL: (609) 896-2255	Tool Touch Detection System
Promess Inc. Brighton, MI TEL: (810) 229-9334	Drill Monitoring System using AE
Prometec Inc. Ann Arbor, MI 48108 TEL: (313) 998-0001	Process Monitor: G90, G100, G110, G200

Table 2. Nominal operating conditions

Grinding wheels	Diameter: 52 mm Concentration: 75 Medium: 10-20 μ m Fine: 2-4 μ m
Tool spindle speed	15000 RPM
Work spindle speed	180 RPM
Grinding in-feed rate	Medium: 50 μ m/min Fine: 7 μ m/min
Total grinding in-feed	Medium: 50 μ m Fine: 12 μ m
Workmaterial	BK7, 40 mm dia. 50 mm thick
Coolant	Challenge 300HT 12.5 gpm; 20 $\pm$ 0.2°C

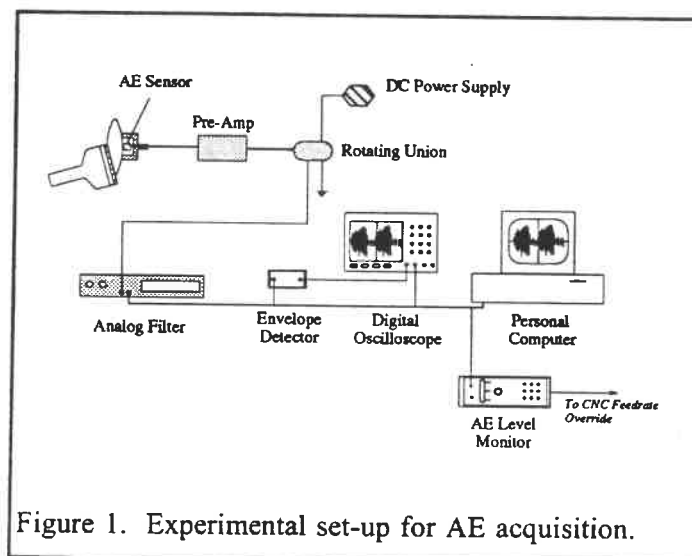


Figure 1. Experimental set-up for AE acquisition.

regarding residual machine deflection, but we believe that it is a suitable reference point for these experiments. There may remain a level of elastic interaction between the tool and the workpiece, leading to a continuing AE signal.⁵

Results

In Fig. 2 is shown a sequence of AE signals measured for different tool-to-work separations, for a single grinding operation. All of the scale lengths are identical for inter-comparison. The length of the abscissa is 6 milliseconds and corresponds to approximately 1.5 revolutions of the tool spindle. Separation is determined by the cumulative z-carriage moves that occurred after a long-term dwell following an earlier grinding operation.

In Fig. 2a is shown two low level signals obtained for large tool-to-workpiece separations. The tool is about 1 mm away from the workpiece in the lower plot and effectively represents the background signal level. The upper plot in (a) shows that the signal increases as the separation closes to 3 microns; we attribute this increase in signal level to turbulent interactions among the tool, the coolant, and the workpiece. In Fig. 2b, the separation is decreased to 1.5 microns, and the rms signal level exhibits a modest increase. As the separation is decreased to 0.5 microns in Fig 2c, AE signal bursts are observed, corresponding to the 'once-per-rev.' period of the tool spindle. In Fig. 2d, as the tool returns to the 'zero' location, determined by the previous dwell operation, the magnitude and duration of these once-per-rev bursts increase. Fig. 2e was obtained during a medium grinding operation with an in-feed of 50 $\mu\text{m}/\text{min}$. Note that substantial signals are observed during the full rotational period of the grinding wheel. Finally, the trace in Fig. 2f shows the long-term dwell (1 minute) of the tool immediately after the grinding operation, which is quite similar to Fig. 2d.

We recorded the sequence of signals from the envelope detector for several experiments similar to that described for Fig. 2, which is shown in Fig. 3a plotted against separation. All of the

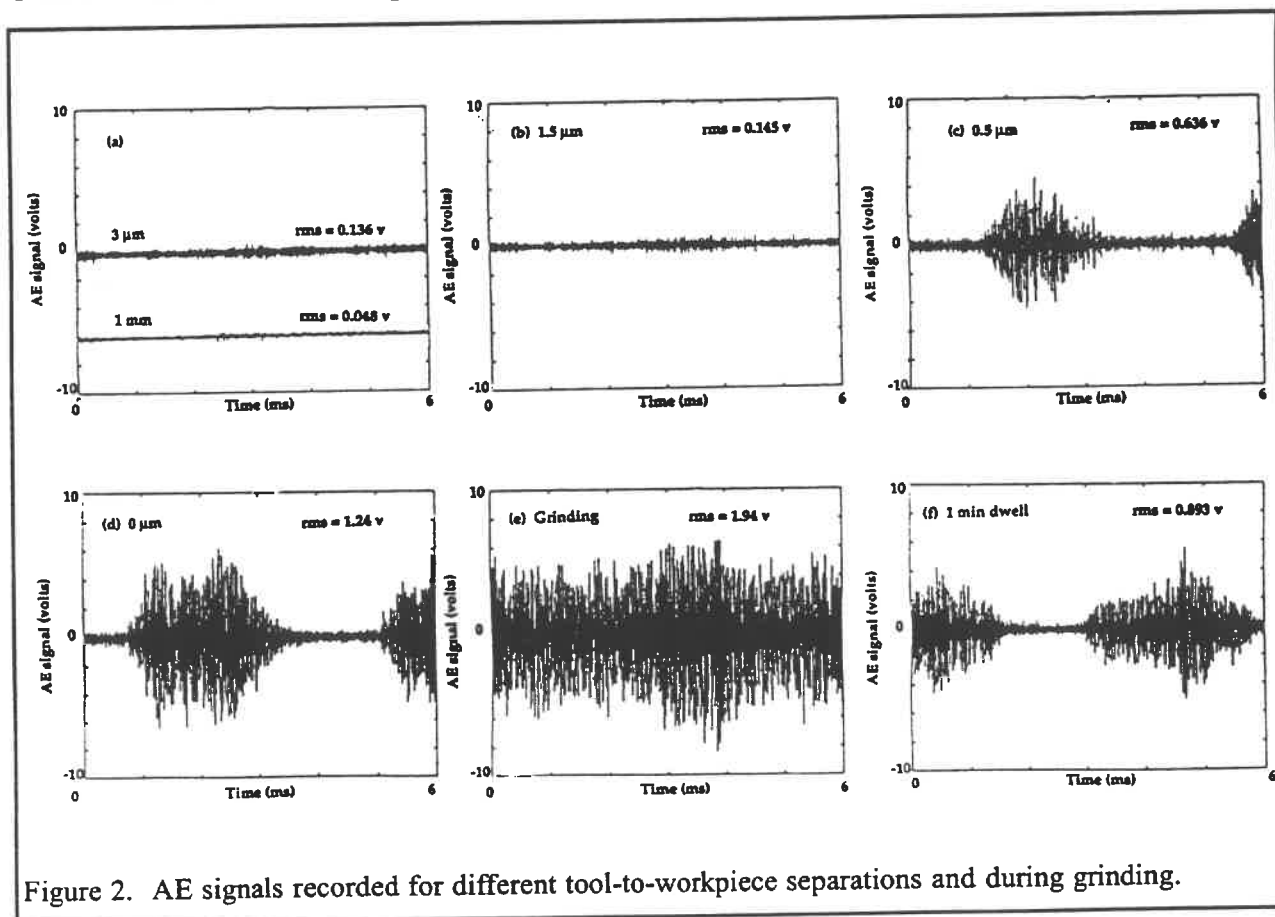
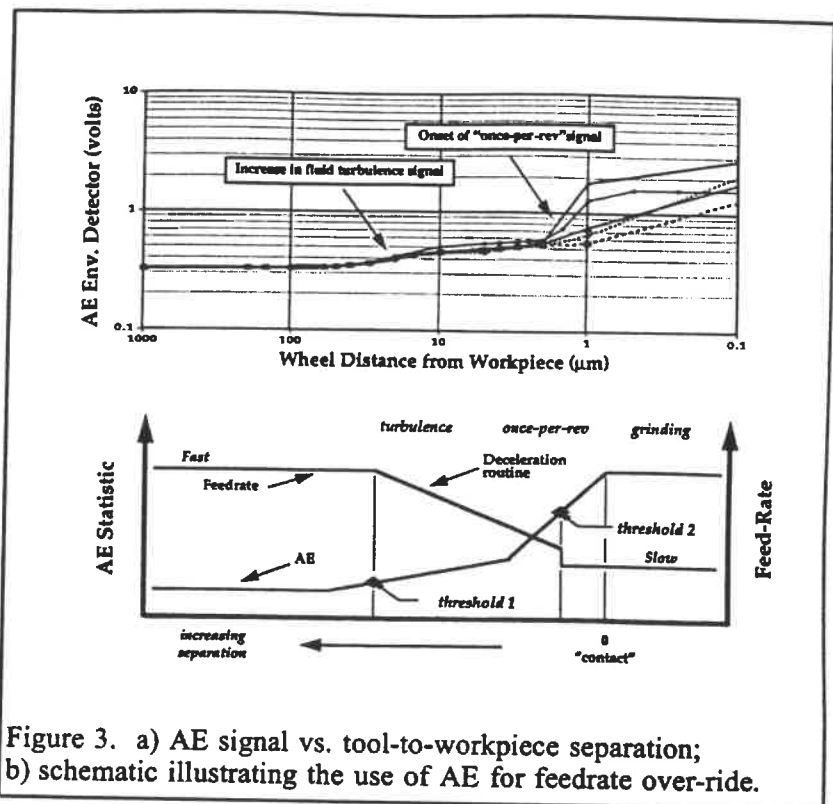


Figure 2. AE signals recorded for different tool-to-workpiece separations and during grinding.

curves measured show a monotonic increase in signal as the separation decreases. Clearly indicated on this plot are the different slope regions corresponding to the turbulent and 'once-per-rev' regions. The turbulent region appears to be identifiable for separations greater than 10 μm . We are also analyzing the source of non-repeatability exhibited in the region labeled as 'once-per-rev'.

In Fig. 3b, we show a potential scheme for using AE information in transitioning from a fast in-feed to a slow in-feed prior to contacting the part. A signal threshold identified in the turbulent regime would trigger the CNC to execute a deceleration routine (shown here as a ramp). A secondary threshold might be invoked in the once-per-rev regime for initiating a faster level of deceleration (shown here as a step). We are currently establishing a feedrate over-ride system on our grinding platform and are assessing the bandwidth and transfer function limitations/requirements for achieving this for various rates of fast in-feed and tool-to-workpiece separations.



Continuing Work

Our immediate goal is to evaluate the use of AE as a feedrate over-ride signal in transitioning from a fast in-feed to the final in-feed. We will also perform similar experiments on the grinding platforms at COM. We will work with optics companies to identify specific requirements for in-feed rates and assess the bandwidths of various controller schemes. We are working to identify optimal statistics for use as control variables, although rms power and envelope signal appear relatively robust. Initial discussions with Prof. Ken Beck at CREOL are leading to further strategies for improving the signal-to-noise ratio of the measurements.

¹ Pollicove, H. M., "The Center for Optics Manufacturing", 1994 Technical Digest for the Optical Fabrication and Testing Workshop, paper OWD1 (1994); Leshne, R. H., "Support for the US precision optics manufacturing base: Center for Optics Manufacturing", Proc. SPIE, vol. 1168, 2-8 (1989).

² Tönshoff, H. K., Zinngrebe, M., Kemmerling, M., "Optimization of internal grinding by microcomputer-based force control", Annals of the CIRP, vol. 35, 293-296 (1986).

³ König, W. and Meyen, H. P., "AE in grinding and dressing: accuracy and reliability", Technical paper MR90-526 from the 4th International Grinding Conference, October 9-11, 1990, Dearborn, Michigan, SME.

⁴ The use of envelope detection is also discussed by 1) reference 3; and 2) Wakuda, M. and Inasaki, I., "Detection of malfunctions in grinding processes", paper presented at the 4th World Meeting on Acoustic Emission and 1st Int'l Conf. on Acoustic Emission in Manufacturing, Boston, September, 1991, ASNT.

⁵ de Oliveira, J. F. G., Dornfeld, D. A., Winter, B., "Dimensional characterization of grinding wheel surface through acoustic emission," Annals of the CIRP, vol. 43(1), 291-294 (1994).

THE DISLOCATION STRUCTURE IN ALUMINA ASSOCIATED WITH ULTRAPRECISION GRINDING

I. Zarudi and L.C. Zhang⁺

Center for Advanced Materials Technology, Department of Mechanical and Mechatronic Engineering,
The University of Sydney, NSW 2006, Australia

1. Introduction

The damage-free grinding processes of hard brittle materials have widely been used in industry for producing electronic and optical components with high surface integrity. It has found that the transition threshold of a material from ductile flow to brittle fracture during the process of material removal plays a central role in the quality control of a ground surface. Material removal in a real ductile mode is micro-cracking free. For metals, such ductile mode is achieved by the easy initiation of multiple slip systems. When cutting ceramics, alumina for instance, strong directional bonding limits the formation of slip systems and causes its brittle behaviour at a temperature below $0.5T_M$ ($T_M=2053\text{ }^\circ\text{C}$ is the melting point of alumina). However, it has not been fairly understood so far what is the scientific principle of the slip system formation when grinding ceramics in a ductile mode at a temperature below $0.5T_M$, and therefore, what is the microscopic mechanism that governs the ductile-brittle transition.

The purpose of this work is to study the mechanism of formation of plastic deformation and subsurface damage of alumina during grinding. As the first insight into the problem, a single-point grinding test is investigated.

2. Method

The material used for study was alumina with average grain diameters of $1\text{ }\mu\text{m}$ and $25\text{ }\mu\text{m}$. The single-point grinding experiments were conducted on a reciprocating sliding machine. (The details of the machine can be found elsewhere [1]). Two types of cutting edges, a blunt conical indenter with a tip radius $100\text{ }\mu\text{m}$ and a sharp Vickers pyramid indenter with an included angle of 136 ° , were used to generate different material removal conditions. The normal loads applied were 0.5 N , 2.5 N and 6 N , respectively. The cutting speed of the cutting edge was 6 mm/s .

The ground surface was examined using scanning electron microscope (SEM). The microstructure of the subsurface of specimens was investigated by transmission electron microscope (TEM). To do this, a ground specimen was first sectioned into thin slices of 1 mm thick, perpendicular to the ground groove. Two slices were glued together in the manner shown in Fig.1. These were thinned mechanically and then by ion milling to achieve a thickness suitable for TEM study. This method of specimen preparation enabled the subsurface damage to be investigated thoroughly in a plane perpendicular to the ground surface. The dislocation structure was examined using the invisibility criterion $\mathbf{g}\cdot\mathbf{b}=0$, where $\mathbf{g}$ is the reflection vector and $\mathbf{b}$ is the Burgers vector.

3. Results and Discussion

3.1 Overall features of the ground surface and subsurface

(1) Effects of depth of cut and cutting edge profiles

The structure of alumina before grinding was almost dislocation free. Extensive plastic deformation took place during the single-point grinding. In the case of coarse-grained alumina ($25\mu\text{m}$ grains) subjected to an external load of 6 N (depth of cut was $10\text{ }\mu\text{m}$), a severely deformed zone, spreaded to a depth of about two grains ($\sim 40\text{ }\mu\text{m}$), was observed near the bottom of the ground groove.

⁺ Author to whom all correspondence should be addressed

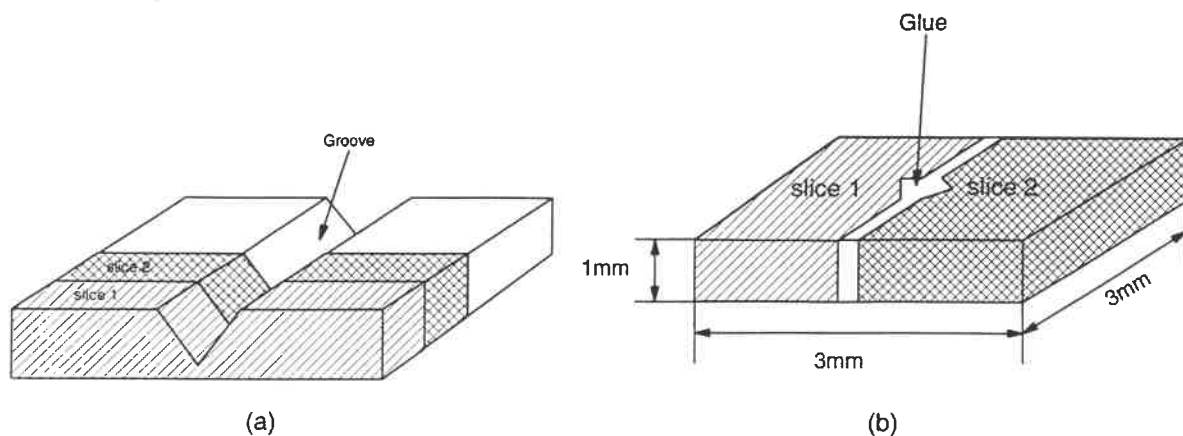


Fig.1. Preparation of specimens for TEM investigations. (a) Cutting thin slices. (b) Gluing two slices together.

This zone contained a high density of inhomogeneously distributed dislocations, deformed twins and a large number of intergranular and transgranular cracks **Fig.2a-b**. The cracks were caused mainly by the intersections between the twins and the twins and grain boundaries. Many grains were fractured and even dislodged during grinding. The groove generated by blunt indenter under the same loading conditions was very smooth. However, this does not imply a pure plastic material removal, because microcracks were found inside grains in the subsurface regions. It therefore follows that an examination on a ground surface cannot guarantee a real ductile mode of material removal. The very different topography of the ground surfaces induced by the different profiles of cutting edges indicates that the shape of the cutting edge has a significant effect on the mechanism of material removal. Cracks decrease remarkably and finally disappear when the depth of cut decreases. The dislocation structure in the subsurface becomes more clear, when the depth of cut becomes smaller. **Fig.2c** demonstrates the regular plane arrays of dislocations and their network in a specimen ground under 0.5 N with the sharp indenter (depth of cut was 0.8 μm).

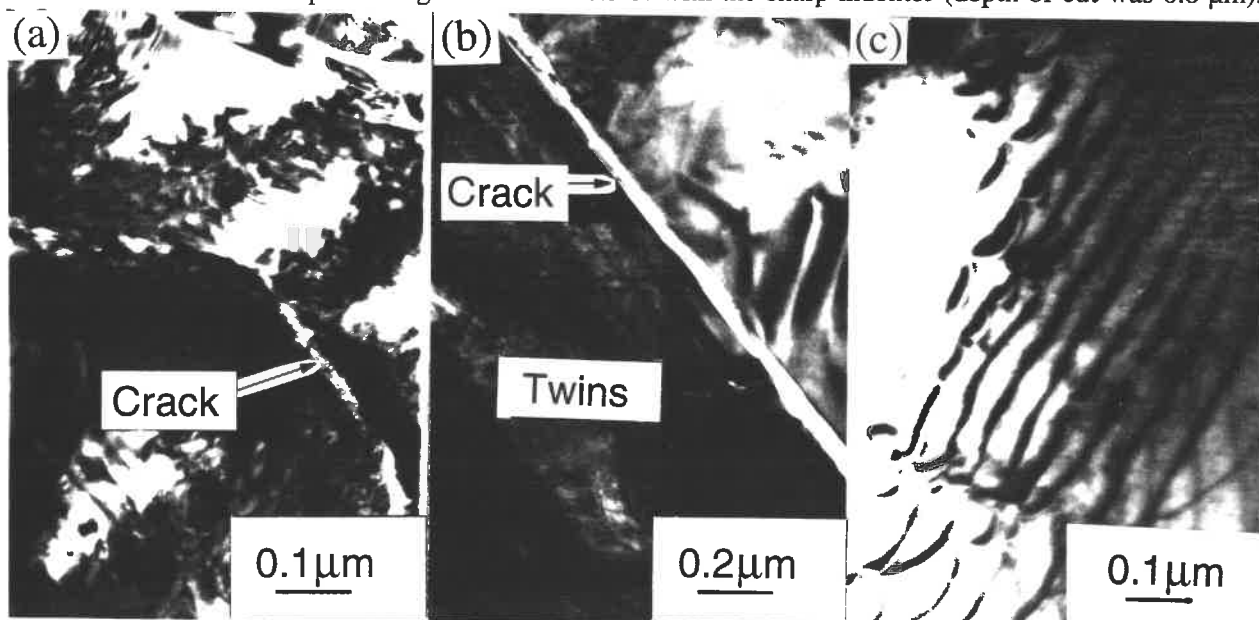


Fig.2. Single-point grinding of the coarse-grained alumina with sharp cutting edge at loads 6 N (a)-(b) and 0.5 N (c). (a) Dislocations and cracks in the subsurface region. (b) Twins and cracks in the subsurface region. (c) Dislocation arrays in the subsurface region.

(2) Effect of grain size

Grain size is an important factor in terms of plastic deformation. For the fine-grained alumina (average grain diameter was $1\text{ }\mu\text{m}$), the grinding with even an external load of 6 N generated smooth grooves (used both blunt and sharp cutting edges), although radial and median cracks were initiated $5\text{--}6\text{ }\mu\text{m}$ below the surface in the case with the sharp cutting edge (Fig.3a). The structure of the plastic zone differed significantly from that in the coarse-grained alumina. The dislocation distribution in the fine-grained alumina was more homogeneous (Fig.3b) and the concentration of twins lower. No intergranular and transgranular cracks were detected. These are favourable from the point of view of crack resistance. Furthermore, the size of the plastic zone in the case of fine-grained alumina was $5\text{--}6\text{ }\mu\text{m}$ which is much smaller than that in the coarse-grained alumina ($40\text{ }\mu\text{m}$). This indicates that initiation of dislocations occurs in the regions $1\text{--}2\text{ }\mu\text{m}$ below the surface and that the grain boundaries limit the development of the dislocations.

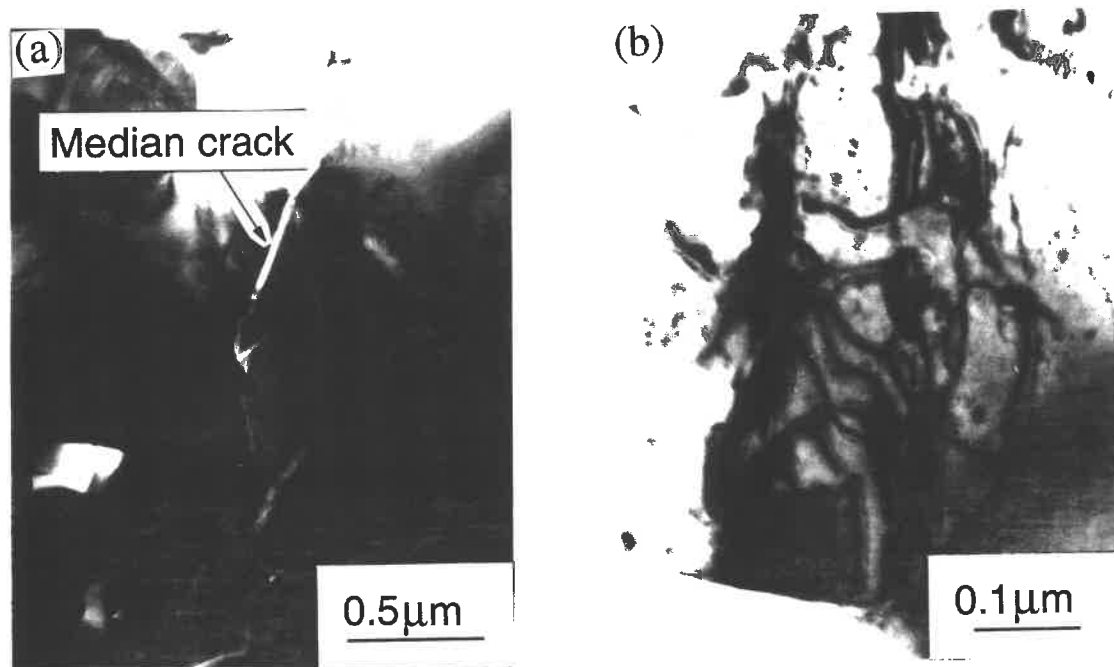


Fig.3 Single-point grinding of fine-grained alumina ($1\text{ }\mu\text{m}$) with the sharp cutting edge at a load of 6 N . (a) A median crack in subsurface. (b) The dislocation structure in subsurface.

3.2 Slip systems

To fully understand the mechanism of the purely plastic material removal when grinding alumina, it is essential to investigate the formation of the possible slip systems activated by the cutting edge. For convenience, let us examine the region with a lower density of dislocations in the ground coarse-grained alumina. Using TEM and the invisibility criterion, the following five slip systems were identified:

- (1) a basal slip system $(0001)1/3\langle 11\bar{2}0 \rangle$,
- (2) a basal slip system $(0001)1/3\langle 1\bar{2}10 \rangle$,
- (3) a prism plane slip system $\{11\bar{2}0\}\langle \bar{1}100 \rangle$,
- (4) a basal twin with its twin plane (0001) and twinning direction $\langle \bar{1}100 \rangle$, and
- (5) a rhombohedral twin with its twin plane (1011) and twinning direction $\langle 10\bar{1}\bar{2} \rangle$.

According to the studies of Groves and Kelly[2] and Chin[3], these 5 systems are all independent. Hence, the Mises criterion of initiating a macroscopic plastic deformation [4] in a polycrystalline body is satisfied. This is one of the microscopic mechanisms that alumina can be ground under a purely ductile mode of material removal.

A further question to answer is how these slip systems were activated during grinding. There are two possible causes for this: (1) the stress field in the workpiece generated by the cutting edge, and (2) the temperature induced by grinding being above the critical initiation temperature of slip and twin systems, i.e., $0.5 T_M$.

We can use the principle of energy conservation to estimate conservatively the maximum temperature rise in the workpiece induced by grinding. Assume that all the input energy through the cutting edge is converted to heat, and that all the heat is conducted into the alumina specimen. Assume further that the heat flux generated by the single-point grinding distributes uniformly in a rectangular area of $-b/2 \leq x \leq b/2$ and $-b \leq y \leq b$ and moves along the workpiece surface with a speed V (Fig.4), where $2b=20 \mu\text{m}$ is the groove width and V is the speed of the cutting edge. Using the well-known Jaeger's model of heat conduction with regard to a moving sources, the maximum temperature rise in the specimen during the single-point grinding is only 20°C , which is far below $0.5 T_M$. Hence, the cause of the above slip and twin system activation was the stress field in the vicinity of the cutting edge. This stress field altered the mechanism of formation of slip and twin systems and made them active even at room temperature.

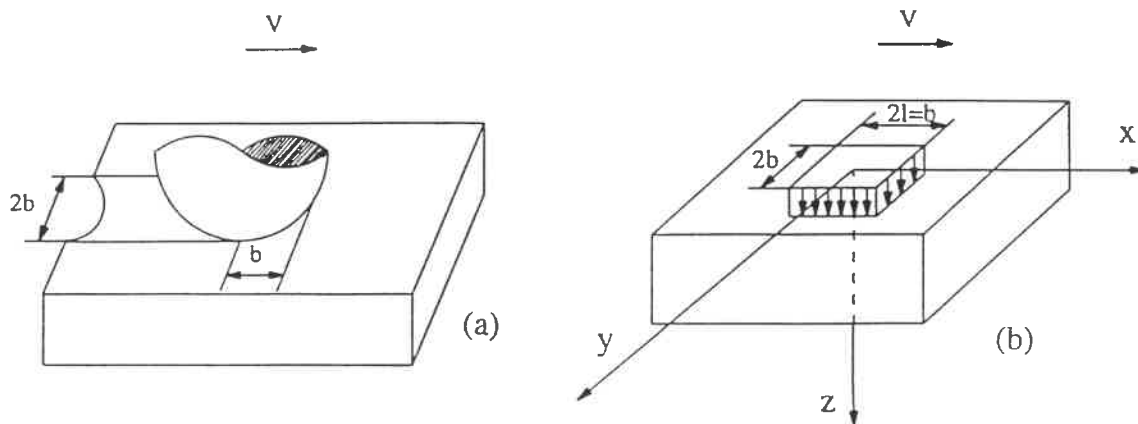


Fig.4 The model for temperature rise calculation. (a) The single-point grinding with blunt cutting edge. (b) The heat source model.

4. Conclusions

- (1) In evaluating the machinability of ceramics in the ductile-regime and estimating the threshold of the brittle-ductile transition, the specific subsurface structure of the material must be considered. Only a surface examination by scanning electron microscopy cannot guarantee a real damage-free grinding. Subsurface damage could be very severe even a machined surface is cracking-free.
- (2) Five independent slip and twin systems have been identified in the deformed zone of subsurface. Because of the activation of these systems, the macroscopic plastic deformation of alumina becomes possible. A real ductile mode of material removal can therefore be achieved by grinding.
- (3) The stress field in the vicinity of the cutting edge altered the mechanism of formation of the slip systems and makes them active even at room temperature.

References

1. A. K. Muhopadhyay, Y.-W. Mai and S. Lathabai, in "Ceramics", edited by M. J. Bannister 2, 910 (CSIRO, Melbourne, 1992).
2. W. Groves and A. Kelly, *Phil. Mag.* **8**, 877 (1963).
3. G. Y. Chin, in "Deformation of ceramics materials", edited by R. C. Brandt and R. E. Tressler, 25 (Plenum Press, New York, 1975).
4. R. Von Mises, *Math. Mech.* **8**, 161, (1928).
5. J. C. Jaeger, *Proc. of the Royal Soc. of NSW* **76**, 204 (1942).

Proposal of High Productivity in Ductile Mode Grinding of Brittle Materials

Akira. Kanai*, Masakazu. Miyashita*, Masakazu. Sato* and Michimasa. Daito**

* Ashikaga Institute of Technology, 268 Omae, Ashikaga, Tochigi 326, Japan

**Nissin Machine Works Ltd., 300 Aritama-nishimachi, Hamamatsu, Shizuoka 431-31, Japan

1. Introduction

The requirements for realizing high productive and ductile mode grinding are discussed in viewpoint of the following items :

- (1) dynamic characteristics of grinding system for realizing stability,
- (2) ductile mode chip removal in chip formation, and
- (3) increase of number of active abrasive grains contributing ductile mode chip removal.

A new diamond wheel satisfying the above three items for high productive and ductile mode grinding is proposed.

2. Dynamic requirements for stabilization of grinding system

Stiffness requirements for stabilization of grinding system and improvement of the motion copying accuracy are

$$K_c < K_{con} < \frac{1}{4\xi} K_m \quad (1)$$

Here, K_c : cutting stiffness of grinding wheel
 K_{con} : contact stiffness of grinding wheel
 K_m : loop stiffness of workpiece - wheel support systems
 ξ : damping ratio of the dominating vibration mode.

Generally, K_c in ductile mode grinding of brittle materials, in particular, productive grinding is high, and K_{con} becomes higher compared with the conventional one due to an increase of number of active abrasive grains. On the other hand, for improving the motion copying accuracy K_{con} is to be higher, as well as static and dynamic loop stiffness of grinding machine. From these considerations, metal bond is most preferable to diamond wheel.

3. Ductile mode grinding and grain depth of cut

The requirement of ductile mode grinding of brittle materials is that the grain depth of cut d_g is no more than the ductile - brittle transition in uncut chip thickness d_c :

$$d_g < d_c \quad (2)$$

The condition is to be satisfied for all chip removal with grinding wheel.

Malkin and others has analyzed geometrically the grain depth of cut and derived the following formula :

$$d_g = 2a \frac{v_w}{v_s} \sqrt{\frac{d}{D_e}} \quad (3)$$

a : spacing between successive active grains.
 v_w : peripheral speed of workpiece.
 v_s : peripheral speed of grinding wheel.
 d : wheel depth of cut.
 D_e : equivalent wheel diameter.
 d_g : grain depth of cut.

The relation between the spacing a and the wheel depth of cut is shown in Figure 1. For example, when $v_s/v_w=100$, $D_e=200$ mm and $a=1$ mm, the wheel depth of cut giving the grain depth of cut larger than $0.1 \mu\text{m}$ is $5 \mu\text{m}$ or more. When $d_c=0.1 \mu\text{m}$, the spacing 1 mm realizes the ductile mode grinding at the wheel depth of cut of $5 \mu\text{m}$.

4. Effect of randomness of protrusion heights of cutting grains

Equation (2) assumes no randomness of protrusion heights of cutting grains and no runout or waviness of grinding wheel. The effect of randomness of the protrusion heights is described as follows :

$$d_g = 2a \frac{v_w}{v_s} \sqrt{\frac{d}{D_e}} + \delta \quad (4)$$

Here, δ : height difference between successive active grains.

Equation (4) means that any active grain with higher protrusion height than the d_c value results no ductile mode grinding. For example, assuming $d_c=0.1 \mu\text{m}$, any active grain with higher protrusion height than $0.1 \mu\text{m}$ generates scratches and does not realize ductile mode grinding. Figure 2⁽¹⁾ shows such scratches on silicon wafers.

Comparing with that the grain depth of cut given by Eq. (3) for $a=1$ mm, $v_s/v_w=100$, $d=1 \mu\text{m}$ and $D_e=200$ mm is 50 nm, it is clear that the randomness of protrusion heights of active grains is a key parameter for realizing the ductile mode grinding .

5. Waviness of active wheel surface and random distribution of protrusion heights of active grains

Runout and waviness of grinding wheel are equivalent in effect to the grain depth of cut and the following discussion treats the waviness of the wheel (ref. Fig. 3).

The envelope of active grains on the wheel is described as follows :

$$h = h_0 \sin 2\pi \frac{x}{\lambda} \quad (5)$$

Here, h_0 : wave amplitude

λ : wave length, x : coordinate along wheel periphery.

The maximum difference of protrusion heights between successive grains $\delta_{\max}$ is

$$\begin{aligned} \delta_{\max} &= \left. \frac{dh}{dx} \right|_{\max} \cdot a \\ &= \frac{2\pi}{\lambda} \cdot h_0 \cdot a \end{aligned} \quad (6)$$

Now, introducing the cut-off length of waviness λ_c , the variation of protrusion heights of the successive cutting grains is defined as random distribution of protrusion heights Δh in the domain of $\lambda < \lambda_c$.

From Eq. (5), in the domain of $\lambda \geq \lambda_c$,

$$\frac{2\pi}{\lambda} \cdot h_0 \cdot a < h_0$$

$$\text{or} \quad \frac{2\pi n}{\lambda} < 1 \quad (7)$$

Therefore,

$$\lambda_c \approx 6a \quad (8)$$

Applying this relation to Eq. (3),

$$d_g = 2a \frac{v_w}{v_s} \sqrt{\frac{d + w_{\max}}{D_e}} + \Delta h \quad (9)$$

Here, $w_{\max}$: maximum runout and/or wave amplitude of grinding wheel surface

6. Diamond wheel for ductile mode grinding

From the considerations mentioned above, the requirements of diamond wheel for realizing a ductile mode grinding of brittle materials are as follows :

(1) protrusion height difference between successive active cutting grains δ is to be :

$$\delta < \delta_c$$

(2) δ is equal to the random height protrusion heights distribution of active abrasive grains Δh which is defined as active grain topography remained after removing waviness of $\lambda \geq \lambda_c$, that is,

$$\delta = \Delta h$$

Therefore, the diamond wheel for ductile mode grinding has to keep the following condition after truing operation and during grinding operation ;

$$\Delta h < d_c$$

And the technical problems to follow the above rule are

- (1) truing accuracy, and
- (2) orientation of abrasive grains and wear characteristics.

Practice of today's strategies for the problems is application of diamond wheel with the following features :

- (1) elastic support of abrasive grains, and/or
- (2) finer abrasive grains.

However, these strategies result the following essential limitations:

- (1) increase of motion copying error,
- (2) low productivity due to the limitation of wheel depth of cut because of finer protrusion heights of abrasive grains, and
- (3) loosening of bonded abrasives, which is absolutely avoided for realizing ductile mode grinding.

Therefore, for realizing ductile mode grinding the diamond wheel has to satisfy the following features :

- (1) high contact stiffness or high abrasive support stiffness,
- (2) high abrasive protrusion,
- (3) no loosening of bonded abrasives, and
- (4) controllability of random protrusion height distribution of within the level of $0.1 \mu\text{m}$.

Fig. 5 is the newly proposed construction of diamond wheel for ductile mode grinding⁽²⁾. Abrasives are monocrystalline pillar diamond and the orientation are aligned.

7. Conclusion

For realization of high productive and ductile mode grinding it is remarked that the random protrusion height distribution of active abrasive grains is to be controlled within the d_c value by truing operation and alignment of crystal orientation of abrasive grains. And diamond wheel with the new construction is proposed.

References

- (1) M. Daito, A. Kanai, M. Miyashita, Observation and Analysis of grinding Scratch Generation around Central Part of Components, Int. Conf. on Optical Fabrication and Testing and Applications of Optical Holography, SPIE, Tokyo, June 5-7 1995.
- (2) A. Kanai, M. Miyashita, M. Daito, J. Hisada, Proposal of diamond wheel with new construction for ductile mode grinding of brittle materials, The 4th Biennial Joint Warwick/Tokyo Nanotechnology Symposium, Univ. of Warwick, Sept. 19-23, 1994.

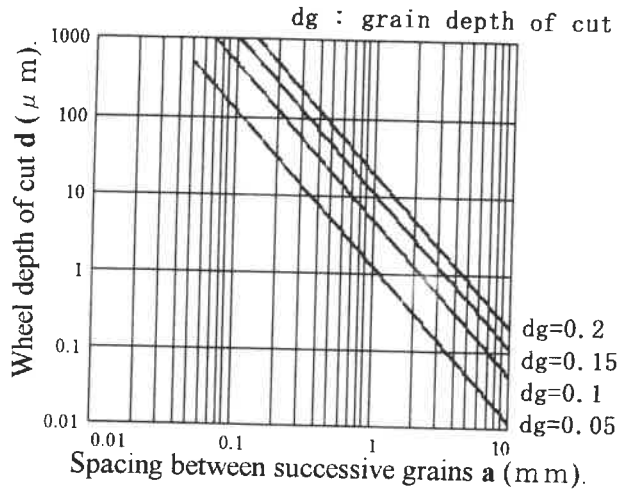


Fig. 1 Wheel depth of cut and grain depth cut.

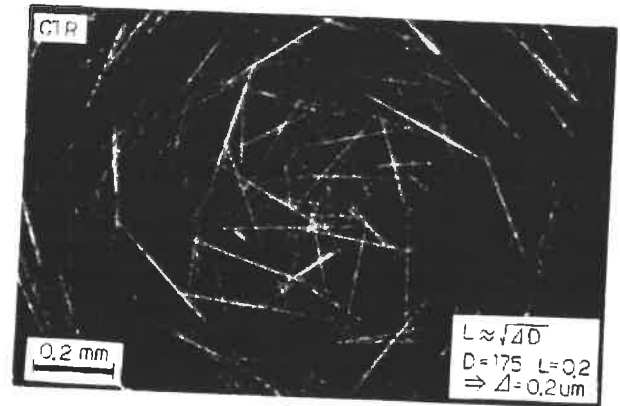


Fig. 2 Scratches on ground silicon wafer.

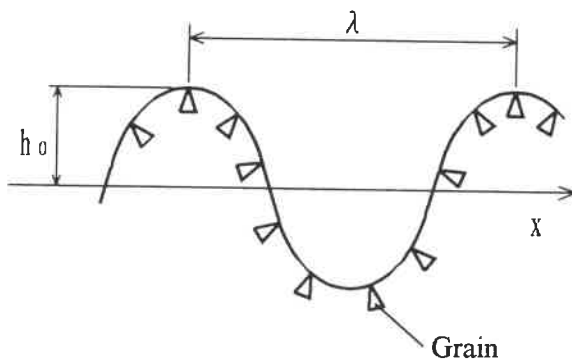


Fig. 3 Waviness of wheel surface.

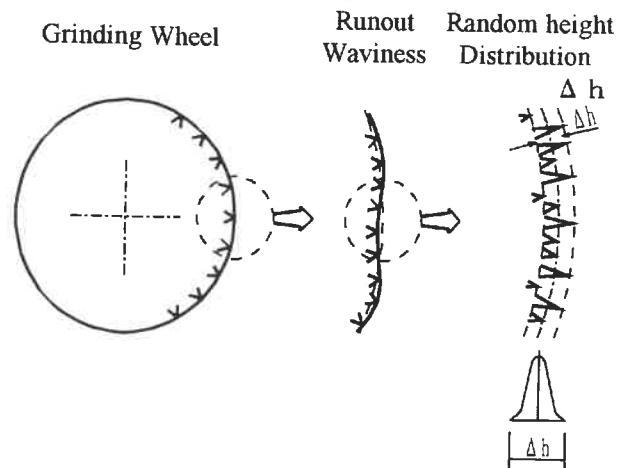


Fig. 4 Protrusion heights of abrasive grains

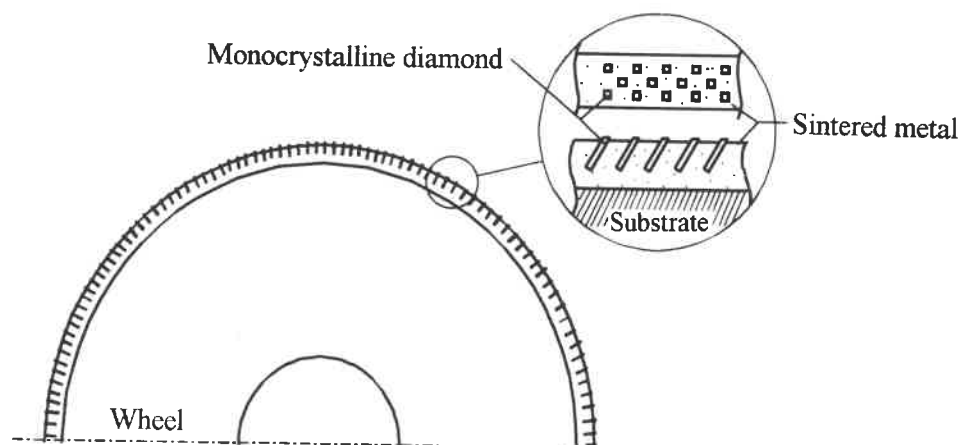


Fig. 5 Diamond wheel composed of monocrystalline diamonds aligned in same orientation.

SHEAR-MODE GRINDING FORCE CRITERIA IN GRINDING WITH SERRATE WHEELS

**Hiroshi Hashimoto, Kenichiro Imai-Kanagawa Institute of
Technology, 1030 Shimo-ogino, Atsugi-shi 243-02 Japan
Tel 0462(41)1211 Fax 0463(93)5440**

Introduction

Shear-mode (or ductile-mode) grinding is accepted as one of the key technologies replacing lapping and polishing processes for brittle materials. This enables standardized processing, based upon the copying principle of the grinding machine.

In designing high precision shear-mode grinding machines to attain high machining accuracy, compliance as a function of grinding force is the most important factor. However, many papers report qualitatively, that an extremely high grinding force is caused by shear-mode grinding compared with brittle-mode grinding. This corresponds with research results based on conventional grinding theory, i.e. 1) the profiles of cutting edges change during grinding, and 2) truing and dressing diversify the surface textures of wheels, all of which cause variation and transition of the grinding force.

This paper describes Shear-mode Grinding Force Criteria (SM'GFC) of various materials and the decreasing method of SM'GFC by the use of serrate wheels.

Definition of Shear-mode Grinding Force Criteria (SM'GFC)

Previous papers^{1),2)} reported shear-mode grinding of BK7 glass using 3000 grit bronze bond diamond wheels ($D=100\text{mm}$). In those experiments the depth of cut was $2\text{ }\mu\text{m}$, the cross feed rate 150 mm/min , and the material removal rate $0.9\text{ mm}^3/\text{min}$. Forces, work and wheel roughness were monitored while repeat passes were made across the work piece. It was shown that:

- * After a number of passes, the normal grinding force reached an approximate steady state;
- * During grinding, the wheel surface roughness and cutting performance were invariant; wheel truing and dressing were not carried out;
- * Surfaces characteristic of shear-mode grinding, i.e. without pits and fractures, were produced.

From these experimental results, shear-mode grinding force criteria were defined as the steady, invariant grinding force which is a repeatable deterministic function of shear-mode grinding conditions including materials and grinding wheels.

Shear-mode Grinding Force Criteria of Various Materials Corresponding to Removal Rates

After preliminary experiments, a 4 mm thick bronze-bonded 8000 grit diamond wheel ($D=100\text{ mm}$) was used to grind 50 mm long, 3 mm thick samples of Zerodur and Pyrex,

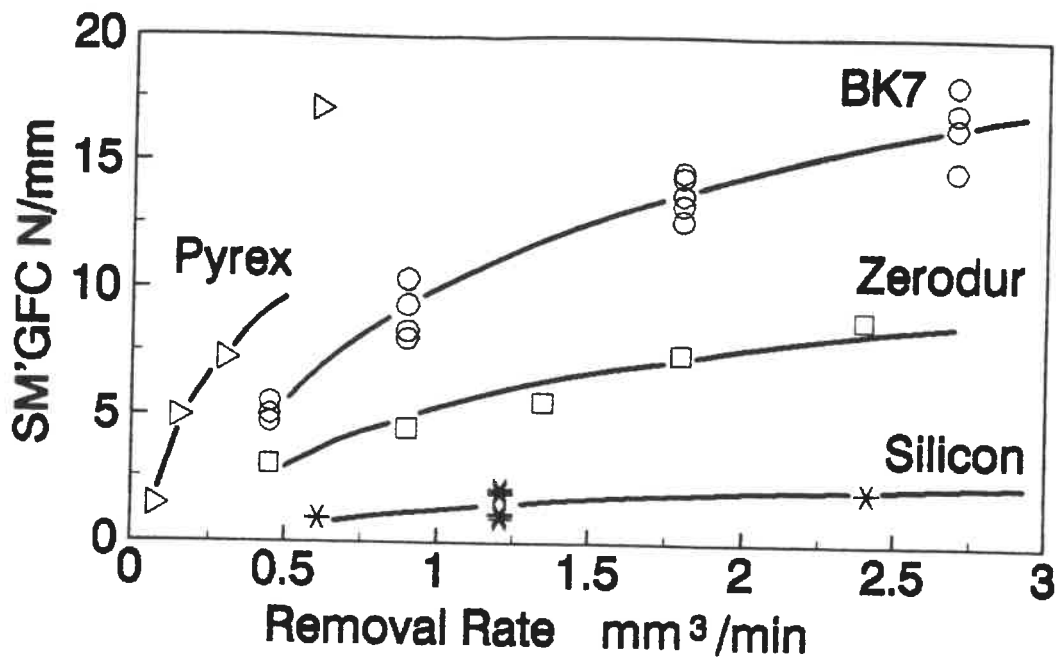


Figure 1 Shear-mode grinding force criteria [SM'GFC] at various removal rate.

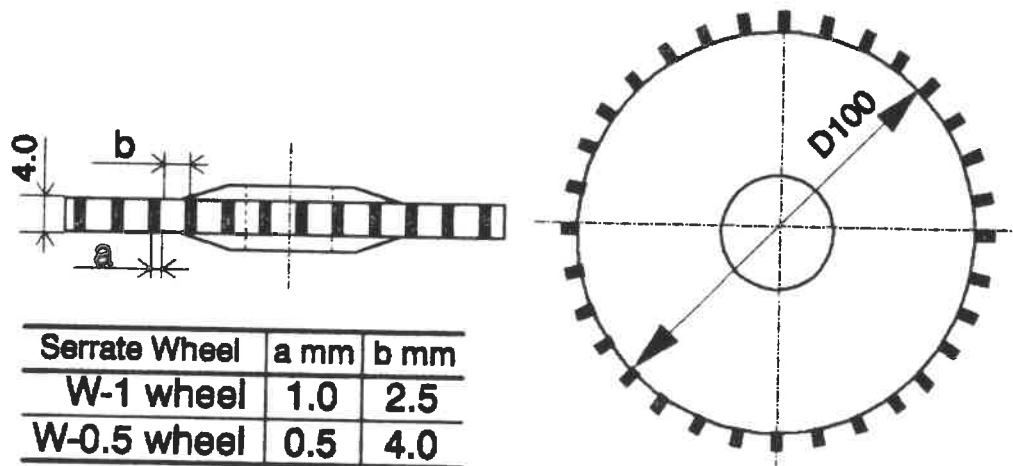


Figure 2 Schematic drawing of Serrate Grinding wheel.

and a 3000 grit wheel ($D=100$ mm) was used for BK7 glass and silicon; an oilbased grinding fluid was used. Wheel speed was 1500 rpm and cross feed rate 200 mm/min for silicon, 150 mm/min for Zerodur and BK7, and 50 mm/min for Pyrex. Grinding forces were measured using a four-axis quartz piezoelectric instrument. Fig. 1 shows SM'GFC of Zerodur, Pyrex, BK7 and silicon at various removal rates. Zerodur has a fairly low SM'GFC in comparison to BK7, but follows the logarithm function curve, that is similar to BK7. Pyrex seems to be on the logarithm curve in the lower removal rate range but deviates from the curve at $0.6 \text{ mm}^3/\text{min}$ and grinding burn was caused at $1.2 \text{ mm}^3/\text{min}$.

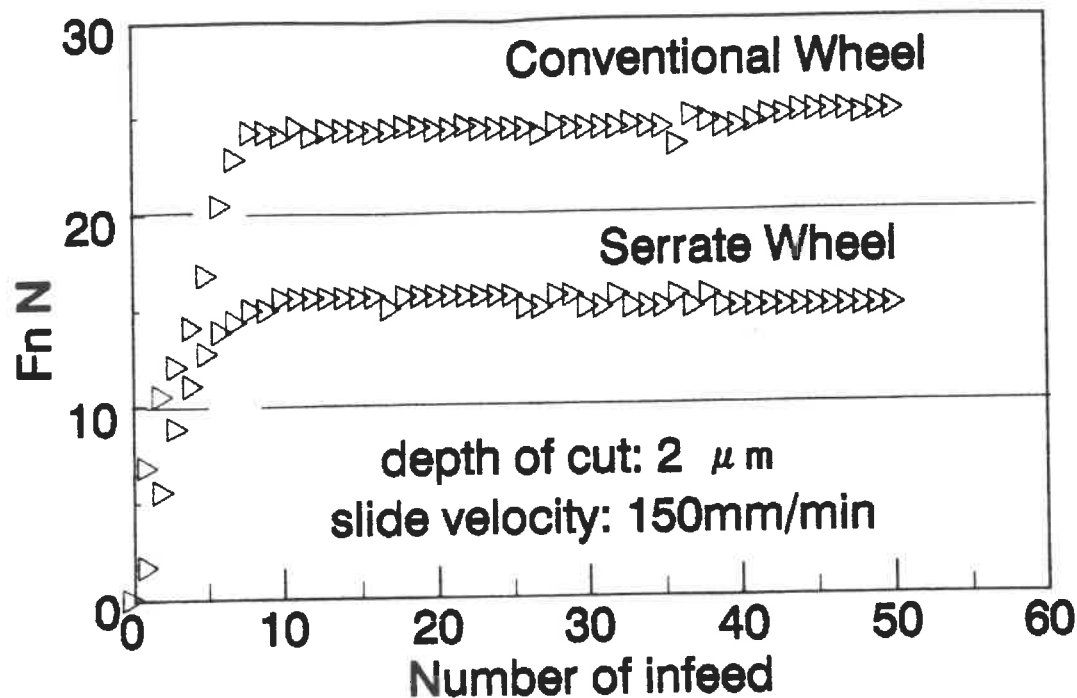


Figure 3 Change of grinding force with W-1 wheel and conventional wheel.

In the case of BK7 grinding, this type of deviation appeared at $4.5 \text{ mm}^3/\text{min}$. The ratio of tangential to normal grinding force F_t/F_n was also steady and constant, and showed about 0.08 for Zerodur and 0.06 for Pyrex, and by contrast, 0.08 for BK7. Silicon showed the lowest SM'GFC among selected test pieces and F_t/F_n was 0.06. But grinding burn occurred at removal rate $4.8 \text{ mm}^3/\text{min}$.

Serrate Wheels and Shear-mode Grinding Criteria

Two 3000 grit diamond serrate wheels (Fig. 2) used for experiments are a W-0.5 wheel with a tooth width of 0.5 mm, and a W-1 wheel with a tooth width of 1 mm. Both wheels have a diameter of 100 mm and a tooth thickness of 4 mm, which are the same as those of conventional wheels. In experiments, grinding conditions were as follows; depth of cut $2 \mu\text{m}$, cross feed rate 150 mm/min and wheel speed 1500 rpm. As shown in Fig. 3, grinding force with a W-1 wheel was invariant, i.e. it showed SM'GFC at 40% lower than that obtained with conventional 3000 grit diamond wheel under the same grinding conditions. Grinding force ratio was found to have increased to F_t/F_n 0.12 compared with F_t/F_n 0.08 of conventional wheels. However, increasing the depth of cut to $4 \mu\text{m}$, the grinding force became unstable and at the depth of cut of $6 \mu\text{m}$, grinding burn was caused. Fig. 4 shows the grinding force obtained with W-0.5 and W-1 wheels under the lower cross feed rate of 50 mm/min. We found that both serrate wheels produce invariant grinding force (SM'GFC).

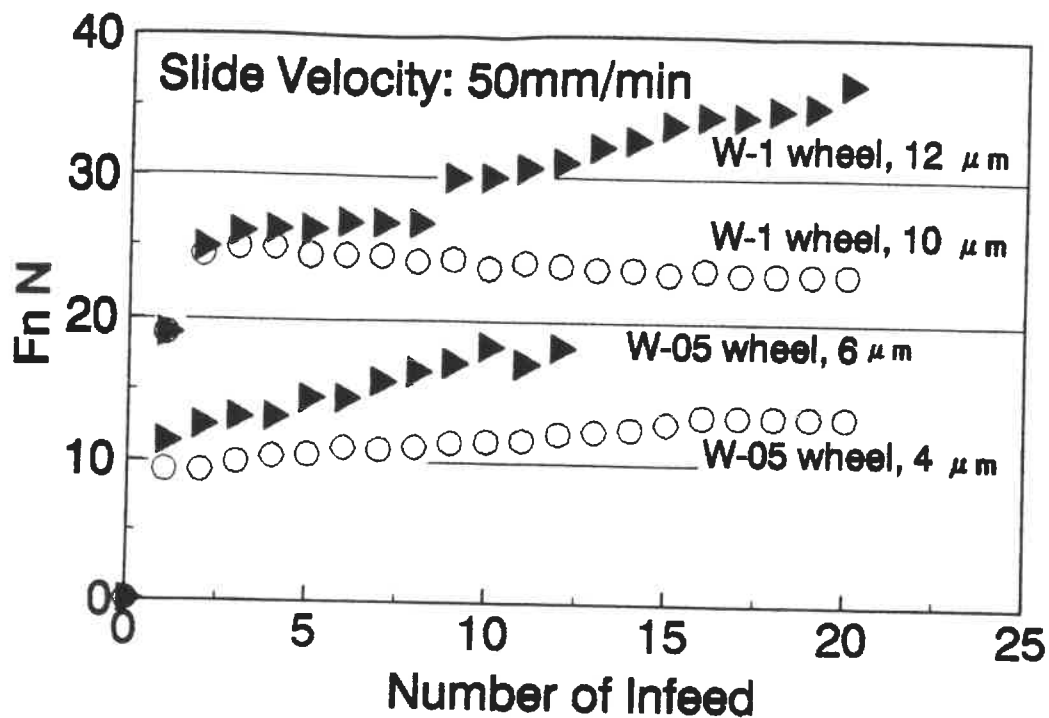


Figure 4 Change of grinding force with serrate wheels.

Comparison of Removal Rate per Grain

Table 1 shows the ratio of removal rates per grain obtained at the largest depth of cut of each wheel, under the above-mentioned grinding condition before grinding burn occurs. From these results, it is assumed that serrate wheels facilitate cooling down of diamond grains by supplying more grinding fluid between wheel and workpiece surface. This effect is also assumed to decrease the normal grinding force and increase the grinding force ratio F_t/F_n as described above.

Table 1 The ratio of removal rate per grain.

Wheels	Ratio
Conventional	1
W-0.5	1.21
W-1.0	1.17

Reference

- 1) K.Imai, H.Hashimoto : Change of Depth of Cut and Damage on Ground Surface in Shear mode Grinding, ASPE Spring Meeting, Tucson AZ, (1993), 95.
- 2) H.Hashimoto, K.Imai : Shear-mode Grinding Criteria of Zerodur and Pyrex, SPIE Proc. Vol.2576, (1995), 9.

High-Efficiency Mirror Finishing of Ceramics with Diamond Wheel

Xu-Jin WANG, Katsuo SYOJI and Tsunemoto KURIYAGAWA

Dept. of Mechatronics & Precision Eng., Tohoku University

Sendai, 980-77, Japan

1. INTRODUCTION

Loose abrasive methods, such as lapping and polishing, can easily produce a very smooth mirror surface using simple finishing machines. However, these methods have a lot of problems in terms of the machining procedures; for instance, the profile error is large, the finishing efficiency is low, and process control is difficult. In order to solve these problems, the authors have developed a new vertical type ultra-precision plane honing machine with an extremely fine grit diamond wheel. Utilizing the developed machine, the high-efficiency mirror finishing of ceramics was carried out successfully^{1) 2)}. In this study, the characteristics of ceramic's high-efficiency mirror finishing using an extremely fine grit diamond wheel was investigated.

2. PLANE HONING MACHINE

The main structure of the developed plane honing machine is shown in Fig. 1. To remove the effect of vibration in the developed ultra-precision plane honing machine, an air bearing was employed on the diamond wheel's spindle, a hydrostatic bearing was also used on the workpieces' spindle. The workpieces were attached to

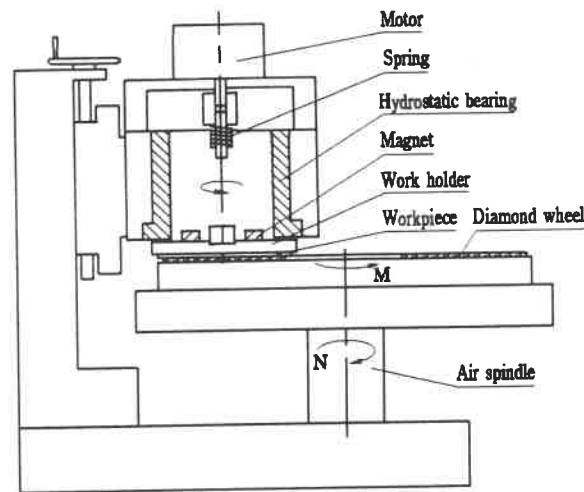


Fig. 1 Ultra-precision plane honing machine

Table 1 Experimental conditions

Grinding wheel:	
Diamond wheel (ϕ 205mm)	① SD1500B100 ② SD5000B100 ③ SD8000V100
Truing wheel	GC1500H (ϕ 90mm)
Workpiece:	
Material	zirconia (ZrO_2)
Size	15mm $\times$ 15mm (6 piece)
Diameter of work holder	ϕ 100mm
Pre-machining method	Lapping ($R_{max} 0.5 \sim 1.6 \mu m$)
Finishing conditions:	
Machining load	30 N
Workpiece speed	100 rpm
Diamond wheel speed	850 rpm
Coolant	Water (15ml/min)

the work holder with glue. The work holder was installed to the workpieces' spindle by magnets. Moreover, in the machine, the constant-load method was used.

The diamond wheel used in the process is a plane type grinding wheel. To prevent the generation of hydro-planing force by the coolant, grooves were cut on the working surface of the diamond wheels. The truer is a cup type GC grinding wheel which was installed on the work holder. Therefore, truing of the diamond wheels is simple. Furthermore, high accuracy and high efficiency of truing can be obtained.

3. EXPERIMENTAL CONDITIONS

Two resinoid-bonded extremely fine grit diamond wheels and a vitrified-bonded extremely fine grit diamond wheel were used in the experiment. The mesh number of the resinoid-bonded wheels are #1500 and #5000. The mesh number of the vitrified-bonded wheel is #8000. Owing to the difference of the wheels' manufacturing process, a spiral groove was cut on the working surface of the resinoid-bonded wheels (Fig. 2), and grooves of concentric circles and radiated lines were cut on the working surface of the vitrified-bonded wheels (Fig. 3). Truing of these diamond wheels was carried out with a cup type #1500 GC grinding wheel.

The material of the workpieces is zirconia (ZrO_2), and those workpieces were lapped with diamond abrasives (Diameter: $40\mu m$) prior to mirror finishing. The workpieces roughness at this stage was $R_{max} 1.5 \sim 1.7\mu m$. The workpieces were attached to the work holder in a concentric

circle using glue (Fig. 4).

The tangential force moment (M) of the machining force was measured from the wheel revolution (N) and the net power consumption (W) of the wheel's air spindle motor.

The experimental conditions are described in Table 1.

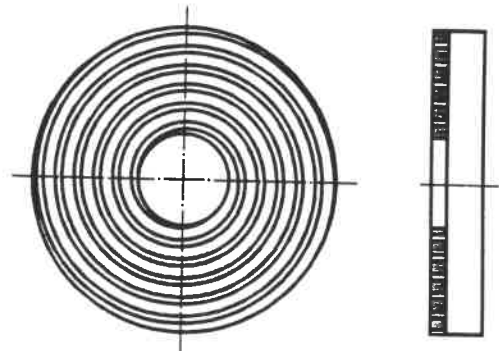


Fig. 2 Resinoid-bonded Diamond wheel

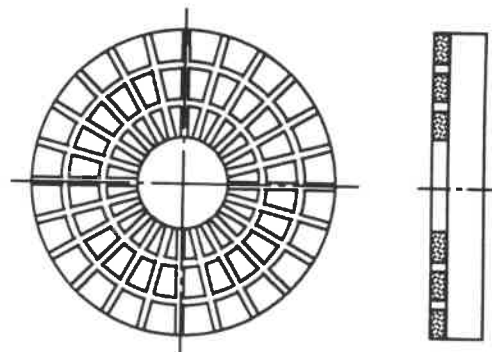


Fig. 3 Vitrified-bonded Diamond wheel

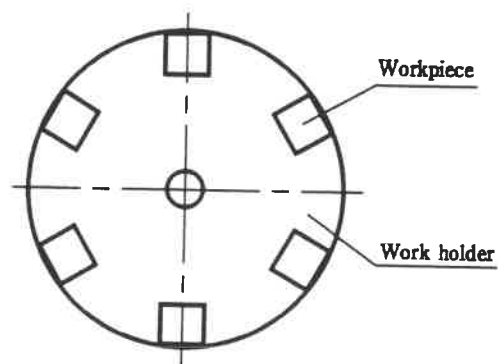


Fig. 4 Workpieces and work holder

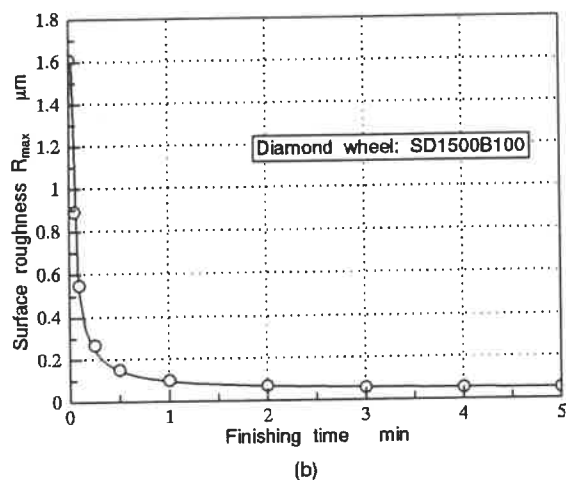
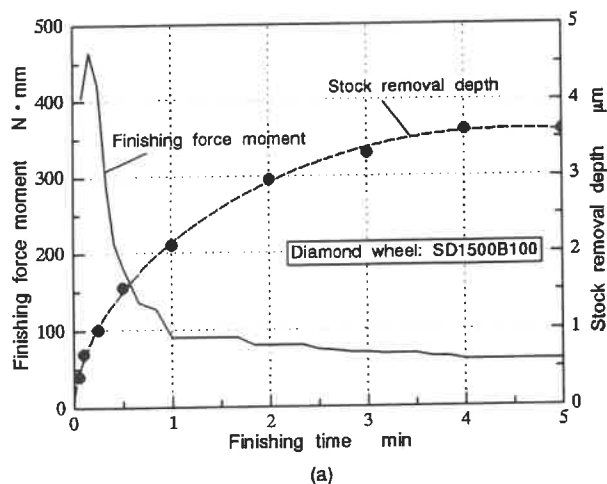


Fig. 5 Finishing force moment, surface roughness and stock removal depth

4. EXPERIMENTAL RESULTS

Fig. 5 shows the changes in the finishing force moment, the surface roughness R_{max} , and the stock removal depth, according to the length of finishing time. The experiment was carried out with a #1500 resinoid-bonded wheel. During the first one minute of finishing, the finishing force moment and the surface roughness decreased rapidly. In the first one minute, an R_{max} $0.1 \mu\text{m}$ mirror surface was obtained. After this, the finishing efficiency deteriorated, and the rate of decrease in the finishing force

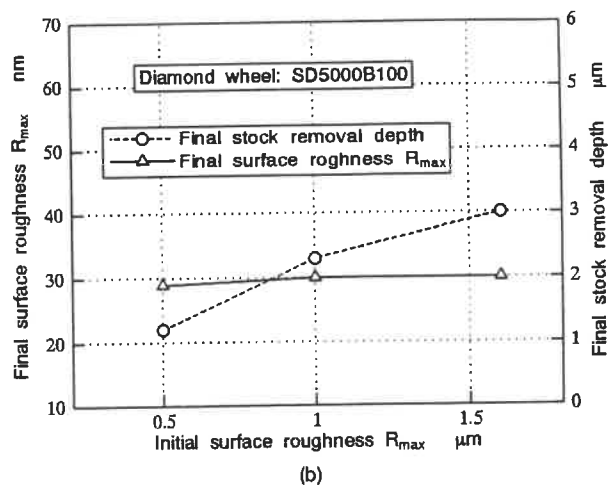
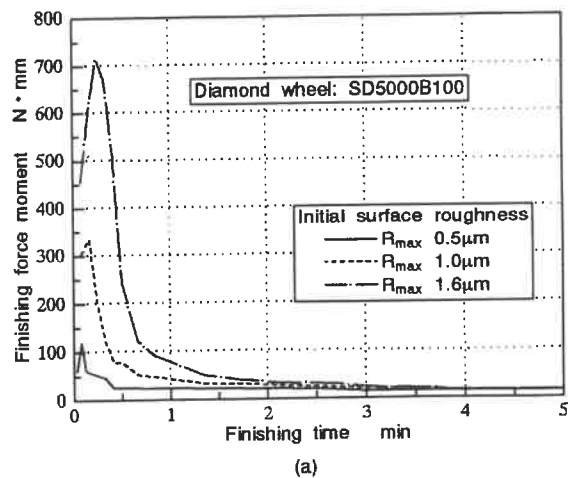


Fig. 6 Effect of initial surface roughness

moment and surface roughness slowed. After 5 minutes, no additional change was observed in the stock removal depth, the finishing force moment, or the surface roughness. The final stock removal depth was $3.6 \mu\text{m}$, and the final surface roughness value was R_{max} 52 nm .

Fig. 6 shows the effect of the initial surface roughness on the finishing force moment, the final surface roughness and the final stock removal depth in the finishing experiment with a #5000 resinoid-bonded wheel. In the early stage of finishing, the larger the initial surface roughness, the

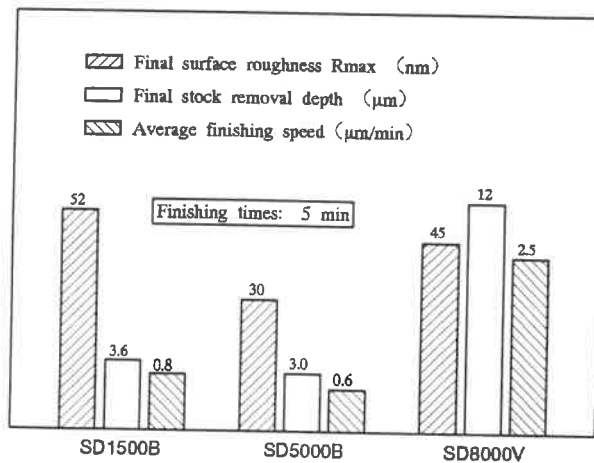


Fig. 7 Effect of diamond wheel's mesh number and bond type

higher the finishing force moment. However, the value of the finishing force moment measured in the final stage was only slightly influenced by the initial surface roughness. When the finishing force moment dropped to a certain value, the rougher the initial surface, the larger the final stock removal depth. Moreover, the final surface roughness values of all workpieces were practically identical.

Fig. 7 shows the results of a comparative experiment among a #1500 resinoid-bonded wheel, a #5000 resinoid-bonded wheel and a #8000 vitrified-bonded wheel operating under the same finishing conditions. The final surface roughness value of the workpieces finished with the #1500 resinoid-bonded wheel (R_{max} 52 nm) was larger than that of the workpieces finished with the #8000 vitrified-bonded wheel (R_{max} 45 nm). However, the final surface roughness value of the workpieces finished with the #5000 resinoid-bonded wheel (R_{max} 30 nm) was less than that of the workpieces finished with the #8000 vitrified-bonded wheel. This indicates that

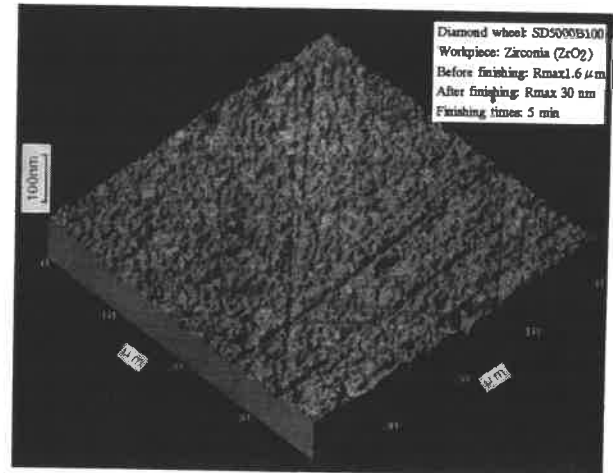


Fig. 8 AFM photograph of workpiece's finished surface

a resinoid-bonded wheel is more suitable for mirror finishing than a vitrified-bonded wheel. The finishing efficiency of the #5000 resinoid-bonded wheel was lower than that of the #1500 resinoid-bonded wheel, but the finishing efficiency of the #8000 vitrified-bonded wheel was three and four times as high as those of the #1500 and #5000 resinoid-bonded wheel respectively.

An AFM photograph of the zirconia (ZrO_2) workpiece finished with the #5000 resinoid-bonded wheel is presented in Fig. 8.

REFERENCES

- 1) X. J. Wang, K. Syoji, H. Nakamura and K. Tanaka: "Studies on Plan Honing Using an Extremely Fine Grit Diamond Wheel", Proceedings of 1994' Spring Annual Meeting of JSPE, (1994) 479.
- 2) X. J. Wang, K. Syoji and T. Kuriyagawa: "High-Efficiency Mirror Finishing of Ceramics with Plan Honing", Proceedings of 1994' Annual Meeting of JSPE, (1994) 95.

Glass Polishing for Optical Communication devices

--Characterization of Slightly Damaged Surface--

NTT Interdisciplinary Research Labs.

Shinsuke Matsui, Humikazu Ohira, Kazuo Matsunaga and Yukio Kikuya

1. INTRODUCTION

In optical communication devices, machine damage to the optical glass surfaces must be accurately characterized and reduced. A communication problem related to machine damaged surfaces is known as the return-loss. This measures the lack of retroreflected light from the optically connected planes ($\text{returnloss (dB)} = -10\log(I_r/I_i)$, I_i is incident light power, I_r is reflected light power. This light destabilizes laser emission, and causes ghosting. Recently we reported that optical indexes of subsurface damaged layers increased by machined residual stress, and their index and thickness data of measured with an ellipsometer could predict the losses quantitatively in the range from 30 to 40 dB¹⁾. In this presentation we intend to establish a polishing process which achieves the high return-loss (40 to 70 dB) demanded recently. We propose a new method of measuring damaged layer so slightly for an ellipsometer. Using this method, topography measurement with an AFM (Atomic Force Microscope) and AFM scratch test, we report the precise structure of surface damage and polishing mechanism concerned to the damage.

2. EXPERIMENTAL METHODS

The optical fiber connector polishing was used for characterization of subsurface damaged layer. Abrasives used for the machining were diamond , cerium dioxide (ceria) and silicon dioxide (silica). For diamond polishing we used diamond lapping films which abrasive diameters were 1 μm and 0.1 μm . For an almost nondamaging process, ceria free abrasive (0.8- μm -diameter)²⁾ and silica fine powder abrasive (0.01- μm -diameter)³⁾ were also used (polisher: politex and cellulosic film, respectively). We measured the reflectance of an optical fiber end plane covered with optical index oil. This reflectance indicates the returnloss. The oil index, sensitive to temperature, can be controlled precisely. Using the principle of two-plane reflection considering interference, the index and the thickness of the damaged layers can be determined

by measuring the reflectance changes with oil indexes. The measurement setup is showed in Fig. 1. Here, the temperature was controlled with 0.1 degree. The oil index varied with -0.00039/degree. This method can measure even very slightly damaged layers because of precise controllability of index and high sensitivity to reflection measurement. The AFM was used for topography measurement of surface near the faibercore.

3. RESULTS

The change in surface during step by step slightly weak aqua-HF etching was measured. The initial surface (return-loss = 40 dB) was prepared with 1- μ m-diameter-diamond abrasive and had a damaged layer with an optical index of 1.54 and a thickness of 20 nm. Fig. 2 shows residual damaged layers after step by step etching. The x-axis indicate the etching depth. The y-axis reveals the damaged layer characteristics (Left axis indicates index and right axis indicates thickness) after etching. This figure reveals that the damaged layer had a high index, 1.54, to almost a 20 nm depth; the index decreased in the deeper section, and the surface etched to 60 nm depth had a 30-nm damaged layer with an index slightly higher than that of the fiber core, 1.452. For the returnloss, the 1.54-high-index layer was dominant. Actually, the returnlosses to the depth of 20nm were same to the values calculated with the mono-layer approximation. Fig. 3 reveals the topography measurements of the etched surfaces. With AFM these etched surfaces had almost same profile to the depth 40 nm etching. Fig. 3a shows the initial surface. Fig. 3b shows the surface etched to the depth of 15nm. However, in the deeper region, the scratches existed at the beginning changed shape, becoming convex bumps (fig. 3c). These facts suggest that underneath the high index layer that determines the amount of return loss, there is a slightly high index region with a structure different from the former.

To get very slightly damaged surfaces, we polished with diamond, SiO₂ and CeO₂ abrasives with particle diameter were 0.1 μ m, 0.8 μ m, and 0.01 μ m, respectively. Fig. 4 shows the characteristics of the damage layer made with these abrasives. The return-losses were 50-60 dB, 50-55 dB and 60-70 dB, respectively. These surfaces had different damaged layers (index (n) =1.52, thickness (d) =5~8 nm with the 0.1- μ m-diamond; n=1.46, d=50 nm with 0.8- μ m-CeO₂ and n=1.453, d=50~100 nm with the 0.01- μ m-SiO₂). The surfaces polished with 0.1- μ m-diameter diamond had thin layers. Their indeces were rather high, and almost same as layers made with 1- μ m-diameter diamond. The damage layers polished with ceria were thick, but their index increase were little (1.46). Figure also reveals that highest returnlosses achieved with silica abrasives. AFM showed that the CeO₂-polished surface had scratches and traces looking like chemical reactions (Fig 5a). However, there were no differences between the surfaces polished with SiO₂ and diamond (Fig. 5b,5c) which were smooth and

scratch-free. On the other hand, These results conclude that even these very weak damaged surfaces have different subsurface damage structures with our method.

4. CONCLUSION

Using a method of measuring slightly damaged surface, we revealed that there is a slightly high index region underneath the high index layer which is polished with $1\text{-}\mu\text{m}$ -diameter diamond, and that even very weak damaged surfaces which are polished with $0.1\text{-}\mu\text{m}$ -diameter diamond, ceria, and silica have different subsurface damage structures. It was also revealed that AFM topography measurement is capable for precise structure of surface damage. The highest returnloss; i. e. weakest damage, was achieved with silica polishing.

Reference

- 1) Ohira, F., Matsui, S., Matsunaga, Saito, T. and Kikuya, Y., Proc. A. S. P. E. 112(1993)
- 2) Orioka, T. and Kasai, T. et. al., J. of JSPE, 31, 11 (1965)900.
- 3) Mtsunaga, K. and Saito, T. et. al., Proceedings of JSPE (Autumn 1989)

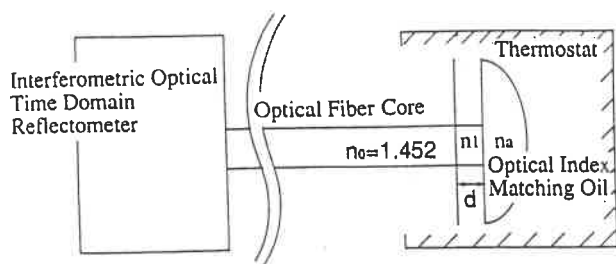


Fig. 1 Schematic diagram of returnloss measurement for precision optical index varying reflectometry method

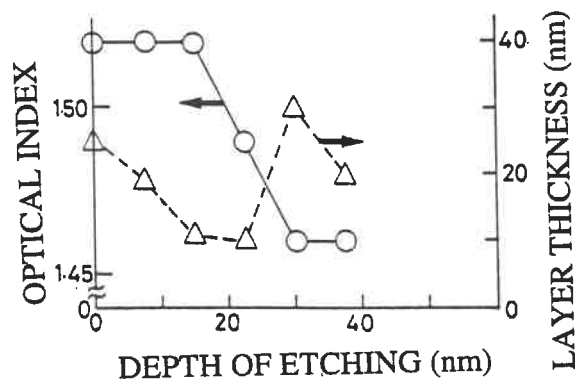


Fig. 2 Change of damaged layer during slightly step by step etching

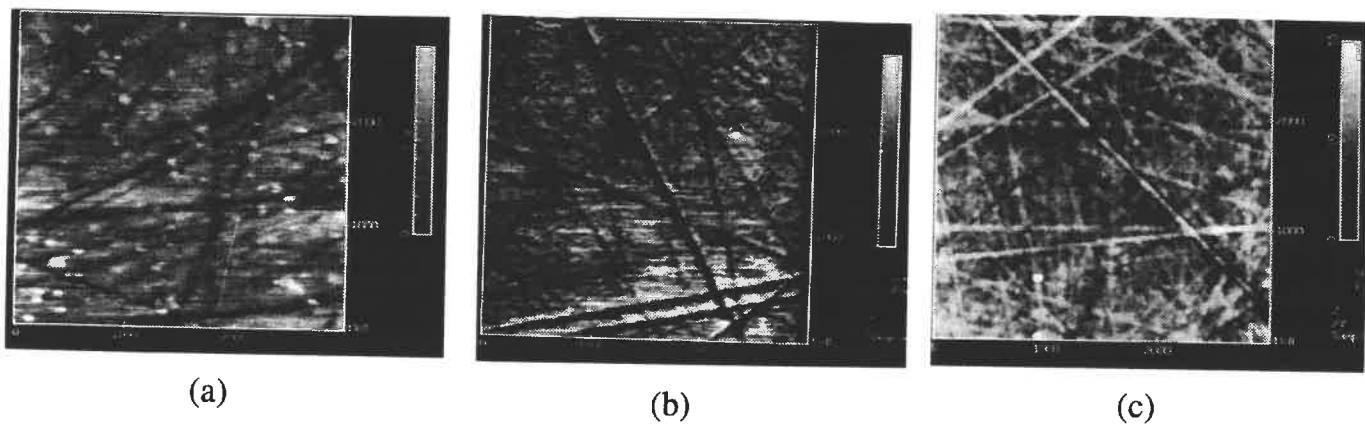


Fig.3 AFM topographical change during step by step etching. (a) initial surface polished with $1\text{-}\mu\text{m}$ -diameter diamond, (b) after 15-nm-depth etching, and (c) after 35-nm-depth etching.

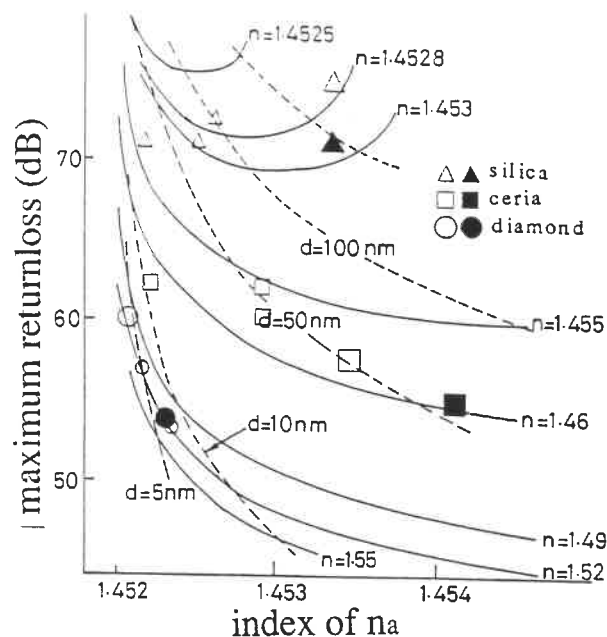


Fig. 4 Relation between maximum returnloss and optical matching oil index at that point with damaged layer index and thickness as a parameter. Solid line represent calculation for damaged layers. Dashed lines indicate iso-thickness. Solid lines indicate iso-index

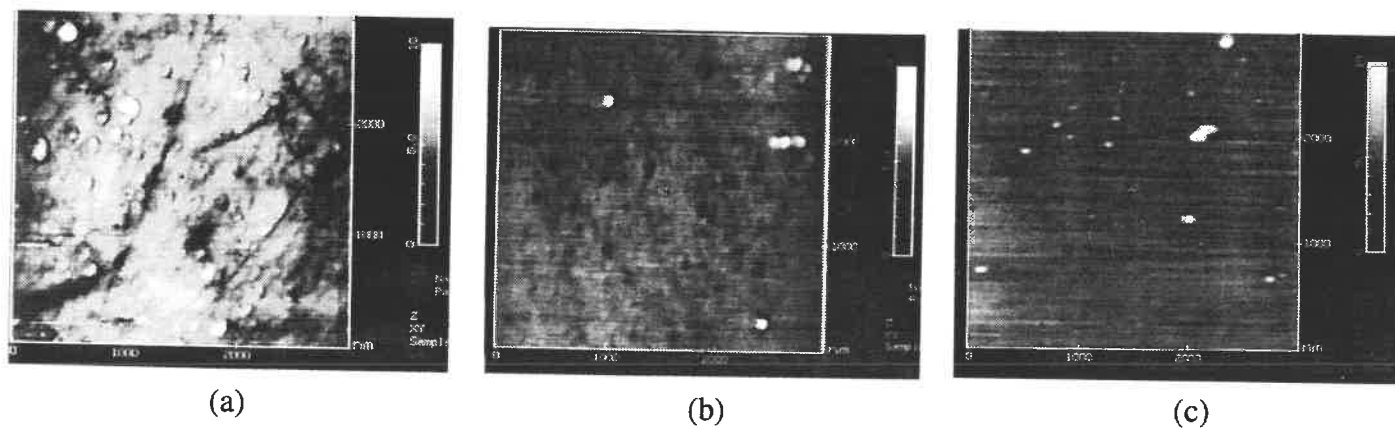


Fig.5 AFM topography (a) with cerium dioxide, (b) Silicon dioxide (c) $0.1\text{-}\mu\text{m}$ -diameter diamond

Grinding of Micro Aspherical Surface

Hirofumi Suzuki, Sunao Kodera, Shigekazu Hara
Mitsubishi Electric Corporation
1-1, Tsukaguchi-honmachi 8-chome, Amagasaki, Hyogo, 661, Japan

and

Katsutoshi Tanaka, Hiroshi Takeda
Toshiba Machine Co. Ltd
2068-3, Ohoka, Numazu, Shizuoka, 410, Japan

1. Introduction

Recently, advanced optical-fiber transmission equipments are being extensively used in various regions and the optical devices are becoming more and more small, such as LD (laser diode) module. The electric information is transmitted to optical signals by the LD module. In the LD module, the semiconductor laser beam spread from the LD chip is condensed to a glass fiber of 10 μ m diameter by a micro aspherical lenses. Accordingly the micro precise aspherical molding dies have been increasingly required for the micro glass lenses.

The sizes of the smallest lens are less than 1 mm in diameter and less than 0.5 mm in the radius curvature. The lens materials are glasses and manufactured by the glass molding method. The workpiece materials of the glass molding dies are ceramics such as WC (tungsten carbide) or SiC (silicon carbide). Therefore they must be ground ultra-precisely by diamond wheel. But in the conventional aspherical grinding method, they can not be ground precisely. Because the conventional grinding head is vertical type and in the case of the concave workpiece of the small radius curvature such as molding die of the micro lens, the diameter of the wheel shaft must be small and the rigidity becomes low.

In this report, precision micro aspherical grinding system was developed and grinding test of a micro molding die was examined. In the grinding test, as a material, WC (tungsten carbide) was ground for the glass molding dies by using a resinoid-bonded diamond wheel of #1200 in the grain size.

2. Grinding machine and method

2.1 Grinding machine

A new grinding method and a grinding system were developed for the micro aspherical surfaces. Fig.1 shows schematic diagram of the micro grinding machine and Fig.2 shows the photograph of the grinding head. The grinding wheel axis is 45 degrees inclined from workpiece rotational axis. The micro wheel is resinoid-bonded diamond wheel and the wheel shape is columnar and its diameter is very small. The wheel is formed to a precise symmetrical column by a single crystal diamond on the machine. The grinding spindle is an air bearing and the maximum rotational rate of grinding wheel is 10,000 rpm. The grinding head is actuated by the simultaneous 2-axis (X,Z) laser feed-back drives.

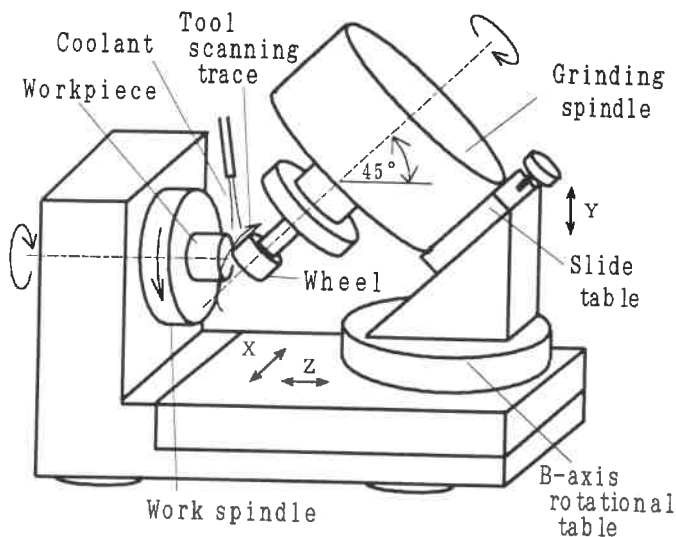


Fig.1 Schematic diagram of micro aspherical grinding machine

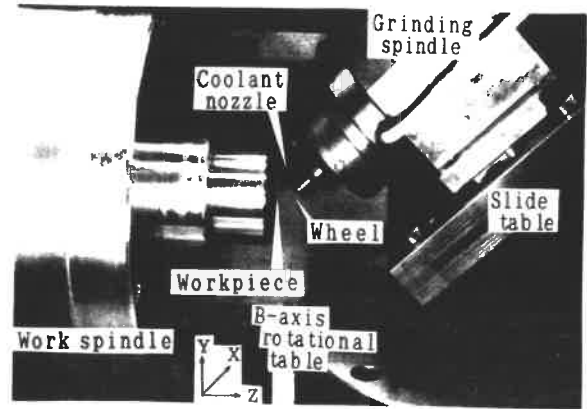


Fig.2 Grinding head

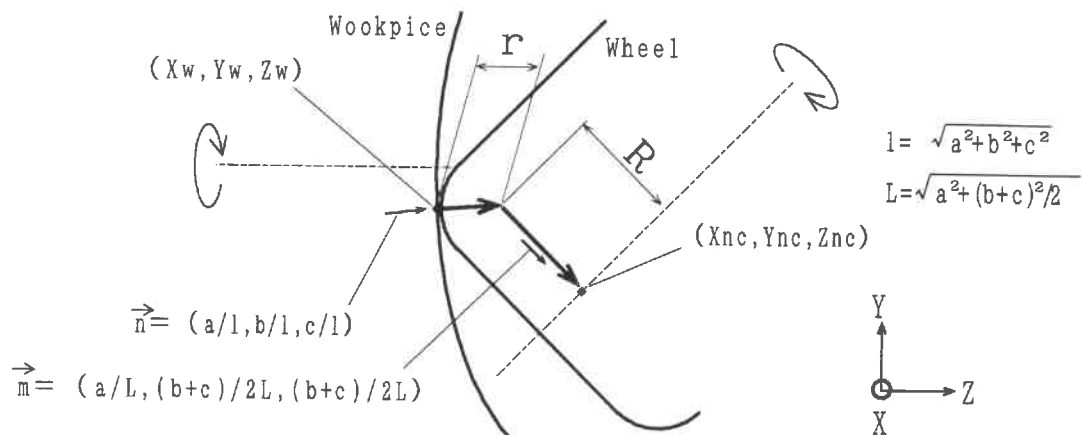


Fig.3 Calculations of tool path

2.2 Tool path

The wheel centroids (X_{nc}, Y_{nc}, Z_{nc}) can be obtained from the coordinate of the grinding point (X_w, Y_w, Z_w) and the normal vector $\vec{n}$ at the grinding point as shown in Fig.3 by the following formulae.

$$\left. \begin{aligned} X_{nc} &= X_w + \frac{a}{l} \cdot r + \frac{a}{L} \cdot R \\ Y_{nc} &= Y_w + \frac{b}{l} \cdot r + \frac{b+c}{2L} \cdot R \\ Z_{nc} &= Z_w + \frac{c}{l} \cdot r + \frac{b+c}{2L} \cdot R \end{aligned} \right\} \quad (1)$$

where, $l = \sqrt{a^2 + b^2 + c^2}$ and $L = \sqrt{a^2 + (b+c)^2}/2$.

Moreover, since this grinder is actuated by 2-axis (X,Z) drives, the centroids (X_{nc}, Z_{nc}) which realize $Y_{nc|X_w=0} = -R/\sqrt{2}$ are calculated numerically by using Newton-Rapson method.

2.3 Grinding process

The tool profiles (outside diameter and tip radius) were at first measured to prepare an NC-program for the primary grinding. The primary ground surface was measured for the distribution of the form deviation profiles. This was to evaluate a reproducible errors such as the errors in tool diameter and tip radius curvature. The NC-program was then edited to compensate the data with regard to the reproducible errors.

3. Grinding experiments

As a workpiece, the molding die shown in Fig.4 was examined and the material was tungsten carbide of ultra-fine particles.

3.1 Grinding conditions

The grinding conditions are shown in Table 1. As a wheel, a resinoid-bonded diamond wheel of #1200 in a grain size was used. The wheel shape was columnar and the tip end was formed in the arc shape as shown in Fig.5. The radius of the tip was reduced as small as possible, 0.1 mm to minimize the grinding force. The truing was performed using a single crystal diamond tool mounted on the workpiece spindle by turning the rotating axis B in 90 degrees.

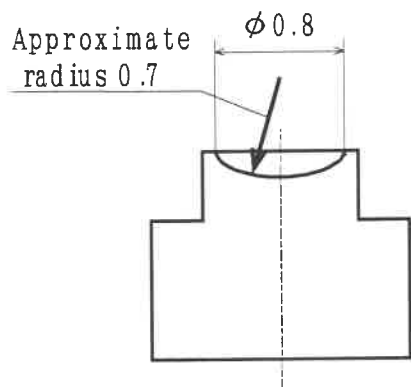


Fig.4 Shape of aspherical die

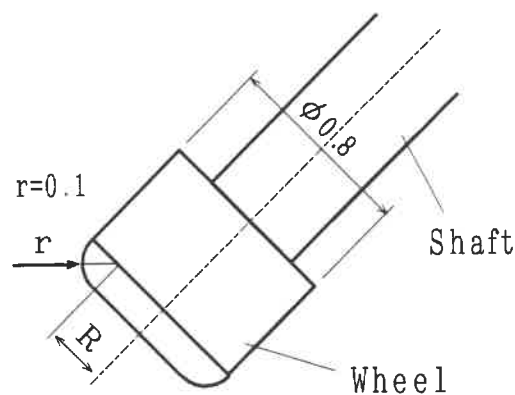


Fig.5 Wheel shape

Table 1 Grinding conditions

Wheel	Resinoid diamond (#1200)
Diameter	$\phi 0.8\text{mm}$
Rotational rate	$10 \times 10^4 \text{ rpm}$
Depth of cut	$0.5 \mu\text{m}$
Coolant	Soluble type
Feed rate	$0.2 \mu\text{m/rev}$

3.2 Grinding results

Fig.6 shows the deviation profile after the primary grinding. The form deviation profiles were measured with Form-Taly Surf. The form accuracy was about $0.4 \mu\text{m}$ P-V. This was caused by the errors of the wheel radius and grinder original position. Fig.7 shows the deviation profile after secondary grinding by correcting the NC program, based on the primary result. The form

accuracy of less than $0.1 \mu\text{m}$ P-V was obtained. Fig.8 shows the surface roughness profile after grinding. The surface roughness of less than $0.03 \mu\text{m}$ Rmax was obtained. Fig.9 shows the Nomarski micrograph of the ground surface. The ductile mode surfaces were obtained.

4. Conclusion

Precision micro aspherical grinding system was developed and grinding test of a micro molding die was examined. In the grinding test, WC (tungsten carbide) was ground for the glass molding dies by using a resinoid-bonded diamond wheel of #1200 in the grain size. The wheel diameter was about 0.8 mm . From the experiment, it was clarified that precision grinding of micro aspherical surfaces can be performed by a resinoid-bonded diamond wheel. By our system, the form accuracy of less than $0.1 \mu\text{m}$ P-V and the surface roughness of less than $0.03 \mu\text{m}$ Rmax were obtained.

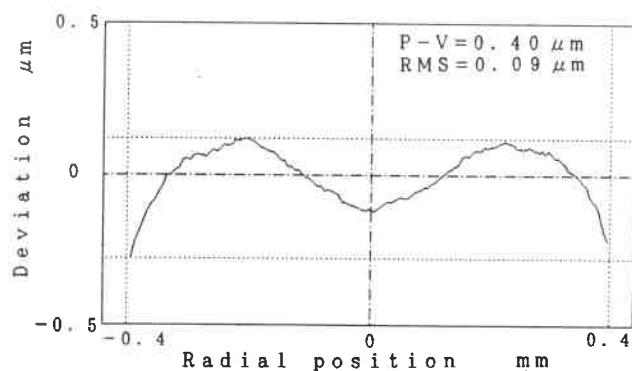


Fig.6 Form deviation profile after first grinding

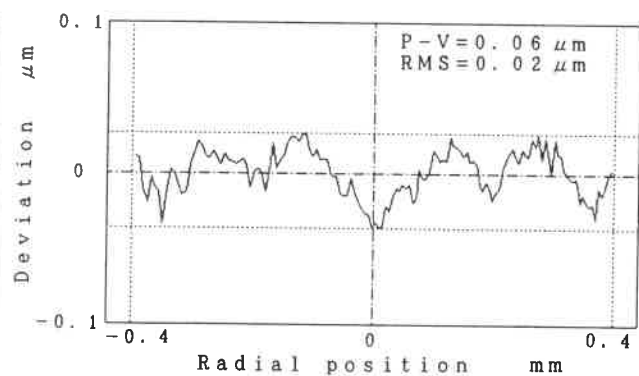


Fig.7 Form deviation profile after secondary grinding

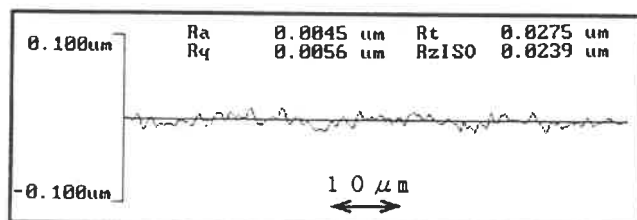


Fig.8 Surface roughness profile after grinding

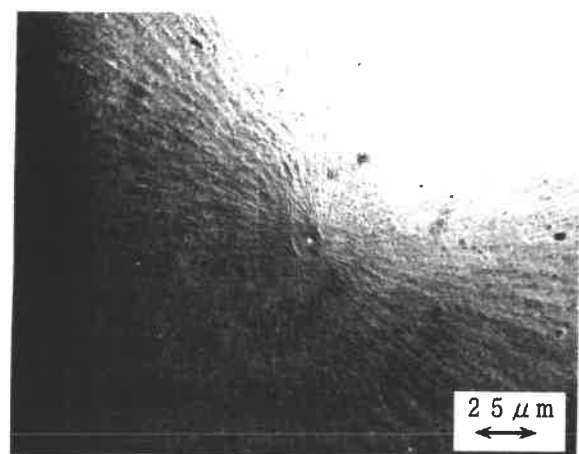


Fig.9 Nomarski micrograph of ground surface

TIME DEPENDENT VOLTAGE & CURRENT BEHAVIOR FOR ELECTROLYTIC IN-PROCESS DRESSING OF BOUND ABRASIVE WHEELS (ELID)

Kevin D. Grobbsky, Doug E. Johnson, and Frank Demarest
Zygo Corporation
Middlefield, CT

Introduction

During the last seven years, a number of publications¹⁻⁴ have appeared describing the merits of electrolytic in-process dressing (ELID) techniques when applied to grinding on hard, brittle materials such as ceramics and glass substrates using fine mesh super-abrasive wheels. ELID grinding, as it is popularly known, has been reported to produce specular surface finishes on these brittle material surfaces, with surface roughnesses on the nanometer (3-6 nm) scale, surface form errors of better than $\lambda/4$, and with little to no subsurface damage. These results were obtained by utilizing commercially available ultraprecision grinding machines coupled with ELID applied as an *in situ* dressing technique for the metal bonded diamond abrasive wheel during the surface grinding process. However, only limited data concerning the underlying physical principles governing the ELID dressing cycle have been published, and to our knowledge, no theoretical explanation of the observed electrical behavior has been published.

In this paper, data is presented for the time dependent nature of the ELID working voltage during the ELID dressing cycle. These temporal changes in the electrical behavior of the dressing process have been measured for several different operating conditions, including input voltages, input currents, d.c. switching times (pulse cycle), and wheel mesh size. We also postulate a possible explanation for the observed voltage versus time dependence.

Underlying Physical Principles

ELID utilizes electrolysis to perform *in situ*, continuous dressing of a fine mesh, fixed abrasive diamond wheel. As illustrated in Figure 1, the fixed abrasive wheel is set up as the positive electrode, the negative electrode is placed in close proximity to the wheel at a known gap distance, and a partially conductive coolant is injected into the gap between the two electrodes. Electric current, slightly below discharge threshold, is then applied to this circuit. The applied current aids in the formation of an oxide layer on the outer (cutting) surface of the wheel. The formation of the oxide layer is believed to occur in two stages⁽⁵⁾. The first stage of this process is a chemical interaction between the coolant and the wheel bond matrix which converts the outer surface of the iron bond into iron hydroxide. The hydroxide layer is then converted into Fe_2O_3 (iron oxide) via electrolysis. The oxide layer, being structurally weaker than the cast iron bond, can therefore be broken down by the grinding process itself. Since current is being constantly applied to the ELID circuit, continuous dressing of the wheel surface is possible.

Theory

Since cast iron is a relatively good conductor while iron oxide is a non-conductor of electricity, the partial conversion of the wheel's cast iron bond into iron oxide through electrolysis effectively alters the resistivity of the wheel as a function of time. One should therefore expect the resistance of the wheel to increase as the electrolysis process continues, with a corresponding increase in the voltage drop across the wheel until a steady state value is obtained.

In order to understand the explicit functional time dependence of the ELID circuit, we modelled the circuit as a simple linear voltage divider similar to that shown in Figure 2. In Figure 2, V_{ps} is the open circuit voltage of the ELID power supply, V_{meas} is the measured voltage output at the circuit junction noted, R_{ps} is the equivalent internal resistance of the power supply, R_c is the resistance contribution of the coolant/gap

combination, and R_w is the resistance of the cast iron bond, diamond impregnated wheel. The instantaneous measured voltage and current at the circuit junction shown can be derived from a simple Ohm's law approach and shown to be given by the expressions:

$$V_{\text{meas}} = V_{\text{ps}} * \{R_w(t) + R_c\} / (R_w(t) + R_c + R_{\text{ps}}) \quad (1a)$$

$$I_{\text{meas}} = V_{\text{meas}} / (R_w(t) + R_c) \quad (1b)$$

Thus, in order to develop a theoretical model for the time dependence of either the measured voltage or current of the ELID circuit, an expression for the time dependence of the wheel resistance must be described. Such an expression can be obtained by using the integral form of Faraday's equation for an electrolytic cell⁽⁴⁾ in combination with Ohm's law, i.e.,

$$R_w(t) = (\eta M / z F \rho_m A^2) * \int (\rho_e) * (I_{\text{meas}}) dt' = K_f \int (\rho_e) * (I_{\text{meas}}) dt' \quad (2)$$

where K_f is a constant for the electrolytic cell. In equation (2), M is the molecular weight of the anode (wheel), I_{meas} is the current flowing in the electrolytic cell circuit, z is the valency of the metal ions disassociating from the anode, η is an empirically determined efficiency factor, A_w is the wheel surface area, ρ_m is the mass density of the cast iron, ρ_e is the instantaneous value of the wheel resistivity, and F is Faraday's constant. If we assume that the oxide layer growth is linear with time (an offshoot of Faraday's equation), it also seems reasonable to assume that the specific functional form for the time dependence of the wheel resistivity should increase linearly with time as a result of the electrolysis process. Under these assumptions, a closed form expression for $R_w(t)$ can be derived. Defining the parameters r and ξ as $r = R_c + R_{\text{ps}}$, and $\xi = V_{\text{ps}} * \rho_0 * K_f$, one obtains (after discarding the negative root solution):

$$R_w(t) = -r + (r^2 + \xi t^2 + A)^{1/2} \quad (3)$$

where A is a constant of integration. The temporal behavior of the measured voltage change can therefore be written as

$$\Delta V_{\text{meas}}(t) = V_{\text{meas}}(t) - V_{\text{meas}}(t=0) \quad (4)$$

where

$$V_{\text{meas}}(t) = V_{\text{ps}} * (-R_{\text{ps}} + (r^2 + \xi t^2 + A)^{1/2}) / (r^2 + \xi t^2 + A)^{1/2} \quad (5)$$

and

$$V_{\text{meas}}(t=0) = V_{\text{ps}} * (-R_{\text{ps}} + (r^2 + A)^{1/2}) / (r^2 + A)^{1/2} \quad (6)$$

Experimental Data

We measured the temporal behavior of the voltage for the ELID dressing cycle. For these tests, a specified mesh size wheel was mounted onto a conventional rotary surface grinder (YGS-16) spindle. Three different mesh size wheels (800, 2000, 6000) were ELID dressed using the ELID-modified equipment. All of the wheels utilized for these experiments had a cast iron bond matrix and an abrasive concentration of 100. The coolant used for the ELID dressing tests was a water-based synthetic coolant with a pH of 9.0 ± 0.2 , and a conductivity of 2.9 ± 0.1 mS/cm. Coolant was applied by both a flood and through-the-spindle arrangement at a rate of $\sim 9 \pm 1$ liters/min.

Prior to initiating the ELID dress cycle, each wheel was dressed and trued using conventional brake dressing techniques to insure that non-oxidized cast iron was exposed. After the wheel was dressed and trued, current was supplied to the ELID electrodes and the temporal electrical behavior (voltage change over time) during the ELID dressing cycle was measured for each wheel. The "raw" data acquired during the ELID dressing cycle was recorded using an Omega chart recorder, then digitized using manual techniques.

Data was generated for each wheel utilizing varying duty cycles (τ = on/off time for the pulse generator). A typical example of the results obtained from these tests is shown in Figure 3. For these experiments, the duty cycle times for the pulse generator were set equal to each other, i.e., $\tau_{\text{on}} = \tau_{\text{off}} = \tau$. The data shown in Figure 3 displays the time dependence of the measured voltage change during ELID dressing for the 800, 2000, and 6000 mesh CI wheel using pulse settings of $\tau = 5 \mu\text{s}$. Although not shown in this paper, the ELID dressing data collected for the other pulse settings indicates similar behavior for each wheel to that shown in Figure 3. The empirical data shown in Figure 3 were then curve fit to equation (4) using commercially available software. The adjustable parameters used for this curve fit were the coolant resistance (R_c), the initial resistivity of the wheel (ρ_0), and the integration constant (A). The results of this best fit analysis are shown in Figure 4. It is seen that the empirical data appears to be adequately described by the analysis suggested by equations (4) through (6).

Summary

The observed changes in voltage as a function of time for each wheel all generally show a slow increase in voltage immediately after the power supply is turned on, followed by a rapid rise in the voltage as a function of time. This rapid increase is followed by an asymptotic approach to some stabilized voltage. However, even though the overall shape of these curves is quite similar, there are several subtle nuances to each of the voltage vs. time curves. Some of these nuances include the small "kink" immediately after the electrolysis process is initiated, the length of time it takes for the wheel to begin to stabilize, and the asymptotic voltage value. For example, based upon the limited data taken to date, it appears that both the rate of the initial rise in ΔV_{meas} as well as the voltage at which the system stabilizes appear to be influenced by the input voltage of the pulse generator (in agreement with equations 5 and 6). In addition, for an identical duty cycle (i.e., fixed value for τ), as the mesh size of the wheel becomes larger (diamonds smaller), the amount of time it takes for the voltage to become stabilized is longer. This suggests that the diamond size and/or number of diamonds in a fixed volume of the wheel influences the time dependent resistivity of the wheel.

Conclusions

A theoretical model of the ELID dressing cycle has been postulated which agrees quite well with empirical measurements. This model suggests that the temporal electrical behavior of the ELID dressing cycle can be approximated as a linear voltage divider with one of the resistors having a simple linear time dependent resistivity.

Acknowledgments

The authors would like to thank Flemming Tinker and William Bacon of Zygo Corporation and Dave Aikens of LLNL for their many useful suggestions. The work presented in this paper was funded by Lawrence Livermore National Laboratories under contract B292121.

References

1. H. Ohmori, "ELID Grinding Techniques for Ultraprecision Mirror Surface Machining", *Int. Jour. Jap. Soc. Prec. Engr.*, vol 26, # 4 pp 273
2. H. Ohmori & T. Nakagawa, "Mirror Surface Grinding of Silicon Wafers with ELID", *Annals of the CIRP*, vol 39 # 1, pp 329
3. T. Bifano and H. Caggiano, "Electrolytic Dressing of Grinding Wheels for Ceramic Machining", presentation given at COM, June 1994
4. E. Welsch, "Electrolytic Dressing of Bronze Bonded Diamond Grinding Wheels", Master's thesis, Boston University (1989)
5. R. Polvani & C. Evans, "In-process Electrochemical Dressing", presentation given at COM, June 1994

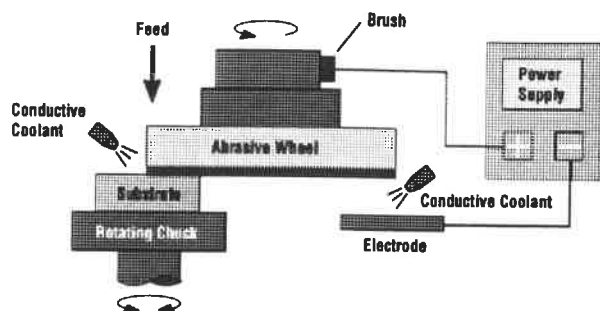


Figure 1
ELID - Electrolytic In-Process Dressing
 An electric current erodes the metal bonded grinding wheel at a controlled rate, exposing fresh diamonds.

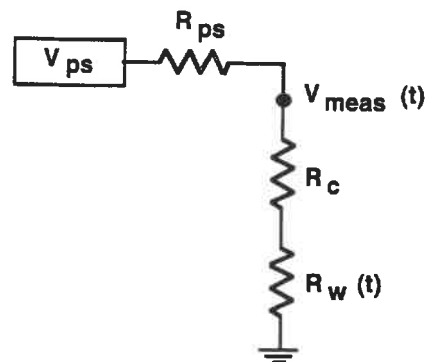


Figure 2
 Linear voltage divider schematic for ELID process.

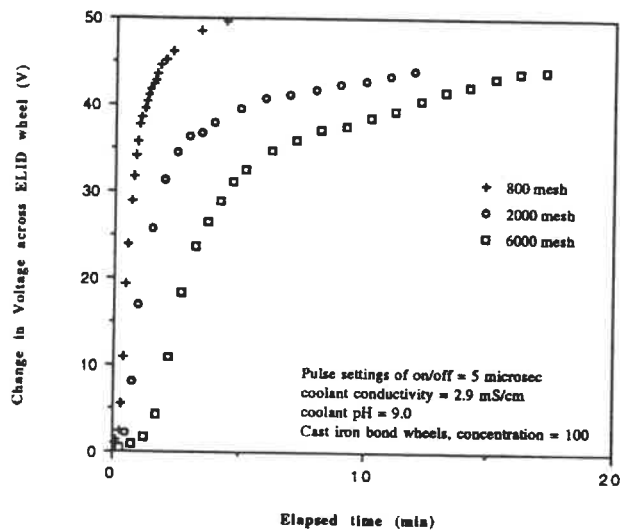


Figure 3
Empirical Measurements of ELID voltage vs. time.

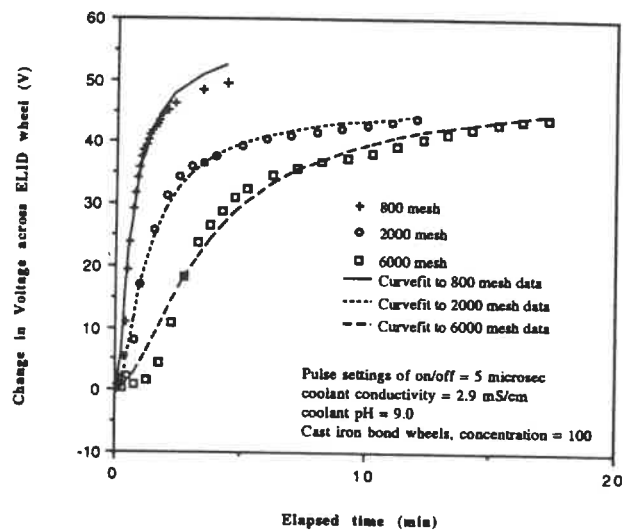


Figure 4
Theoretical Curve fit using equation (4) and data from Figure 3 of ELID voltage vs. time.

ELECTROLYTIC DRESSING OF BRONZE BONDED DIAMOND GRINDING WHEELS

Helen Caggiano, Thomas Bifano
Precision Engineering Research Laboratory
Boston University
Boston, MA

ABSTRACT

In electrolytic in-process dressing (ELID), a metal-bonded diamond-abrasive grinding wheel is dressed as a result of anodic dissolution. In this paper we describe experiments to evaluate the potential for ELID on bronze-bonded wheels in fixed-load and fixed-rate grinding applications. Appropriate ELID conditions were determined for a two bronze-bonded diamond grinding wheels used to machine silicon carbide and fused silica. Some of the important factors controlling the rate of dressing by ELID (rate of film growth, rate of film wear and rate of diamond wear) and their combined effects were measured and analyzed.

INTRODUCTION

Fixed abrasive diamond grinding is the predominant method of machining ceramic and glass materials. By virtue of their hardness, such materials rapidly wear out the diamond cutting edges in the grinding wheel. With some combinations of grinding wheel, workpiece material, and grinding conditions, the grinding wheel bond is eroded by the workpiece at a rate sufficient to continuously expose new, sharp diamond grains while eliminating old, dull grains. Unfortunately, such "self-dressing" processes are rare; typically, as the diamonds wear out, the grinding process becomes progressively less efficient. Worn diamonds in the grinding wheel usually lead to increased grinding force, increased grinding damage, and increased specific grinding energy¹. Figure 1 illustrates an example of the increase in specific grinding energy as a function of the total volume of material removed from a workpiece for an ultra-precision grinding operation on a ceramic workpiece [1].

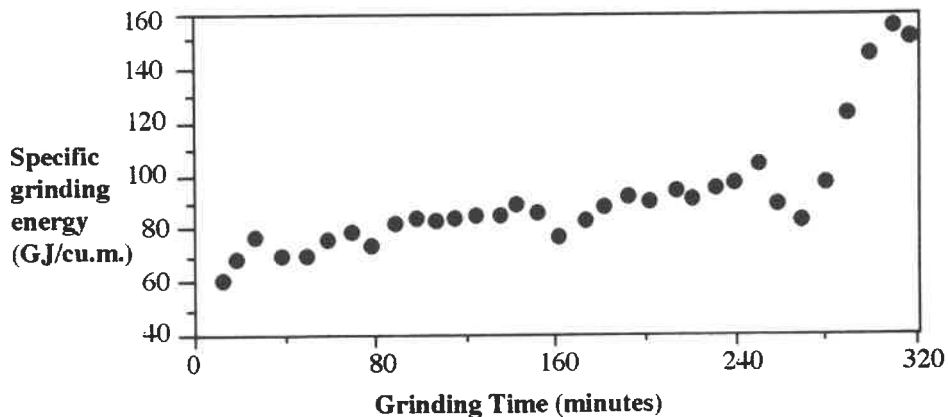


Figure 1. Specific grinding energy as a function of total grinding time in a blanchard-style micro-grinding operation on CVD Silicon Carbide. The sharp increase in specific energy is a result of diamond wear [1].

In this process, a $4-8\mu\text{m}$ grit bronze-bonded diamond grinding wheel was initially dressed and was then used to blanchard-grind silicon carbide on an ultraprecision grinding apparatus. A total of 4 cubic mm were removed from the workpiece in the 320 minute test. Scanning electron microscope (SEM) photographs of the surface of the wheel showed dulling of the diamond grains, loading of the wheel with grinding swarf, and loss of diamonds from the wheel surface.

When wheel performance deteriorates to such a degree it is necessary to perform a dressing operation to remove worn diamonds and grinding swarf and to expose an underlying layer of sharp diamond edges. One such dressing method that has been the subject of research for several years is ELID. ELID has been found to be a useful dressing process for ultraprecision grinding of ceramics and glasses, particularly with iron-bonded diamond grinding wheels [2,3,4]. Previously, we reported on a series of phenomenological studies directed towards understanding the dynamics of electrolysis and oxide film formation on bronze

¹specific grinding energy is defined as the ratio of grinding energy input to material removal volume.

wheels while they were *not* in contact with a workpiece. We found that the oxide film grew both into and out of the original wheel surface. Results from one such set of experiments are shown in Figure 2. The graph on the left depicts the oxide height above the original wheel surface, while the graph on the right depicts the oxide infiltration into the bronze surface.

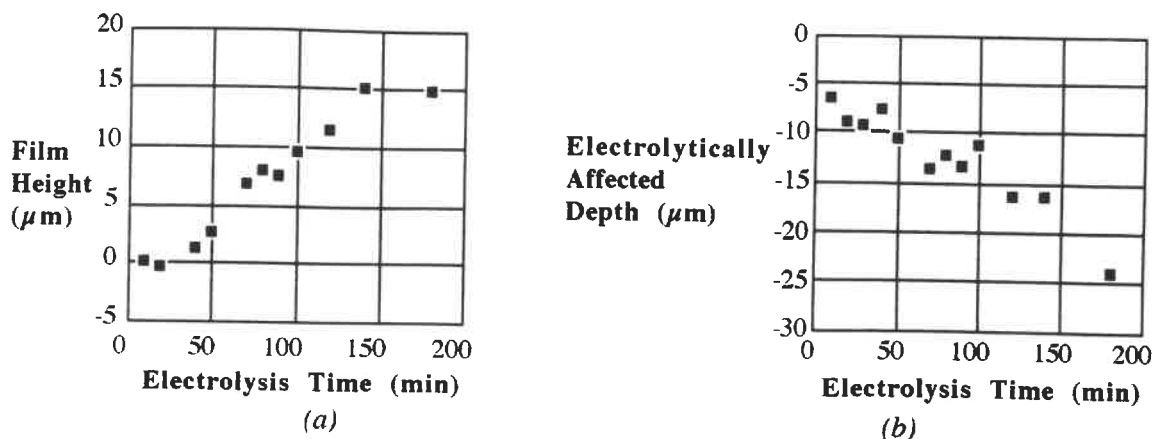


Figure 2 Oxide film growth as a function of time. (a) Film height above original surface. (b) Film depth below the original surface [5]. Each graph indicates a slope of $\sim 0.1 \mu\text{m}/\text{minute}$, suggesting a combined film thickness growth rate of $\sim 0.2 \mu\text{m}/\text{minute}$.

Eventually, the oxide film grows thick enough to obscure diamonds that originally protruded from the surface. However, in an actual grinding process, it would be expected that the erosion rate of the oxide film due to abrasion with the workpiece would be greater than the rate of oxide film generation. In that case, the dressing rate or rate of wheel recession would be equal to the rate at which the film grows into the wheel. The experimental rate of bond erosion is then equivalent to the rate of change of the electrolytically affected depth, which can be estimated from the data in Figure 2 at about $0.1 \mu\text{m}/\text{minute}$. This is within $\sim 20\%$ of the value predicted by Faraday's laws of electrolysis for these conditions.

For ELID to continually expose sharp diamond edges in an actual grinding process, the film that results from electrolysis must be continuously eroded. To explore the dynamics of this process, a constant force grinding apparatus was constructed. Figure 3 illustrates a schematic of this constant force grinding apparatus. Included in the Figure is a summary of the parameters used in these experiments.

- Bronze cup wheel
- 100 mm diameter, 6 mm rim
- 10-20 μm diamond, 75 conc.
- 1 m/s wheel speed
- 5 N constant-force grinding
- HP SiC workpiece
- 6 mm x 9 mm contact area
- 250 μm electrode gap
- 0.0002 M NaCl coolant (Low Current)
- 0.002 M NaCl coolant (High Current)
- 150 Volts DC or pulsed DC (1000 Hz, 50% Duty Cycle)

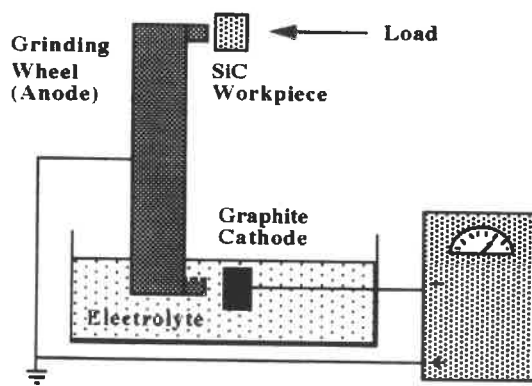


Figure 3 Constant force ELID grinding apparatus

Samples of hot-pressed silicon carbide were used as workpieces and a constant force of 5N was applied during grinding. This normal force was selected because it is close the force levels previously measured in a successful ultra-precision grinding process with this workpiece material, grinding geometry, and wheel [1]. The wheel speed chosen was lower than that normally used in grinding, to avoid coolant splashing and to ensure that the inter-electrode gap was completely filled with electrolyte. Grinding tests were conducted both with and without ELID. Voltage, current, coolant conductivity, temperature, and grinding rate were measured during the experiments. Electrolysis current was adjusted by varying electrolyte conductivity.

The grinding process efficiency can be measured by observing the cumulative material removal volume or cumulative depth-of-removal². If the wheel becomes dull or loaded with debris, or if an electrolytically-induced film obscures the diamond grains, the material removal rate will decline, resulting in an asymptotically limited cumulative material removal depth. If the wheel is successfully dressed in-process, the material-removal rate will reach a steady-state value, and cumulative material removal depth will increase linearly with grinding time. Figure 4 summarizes the constant force grinding results.

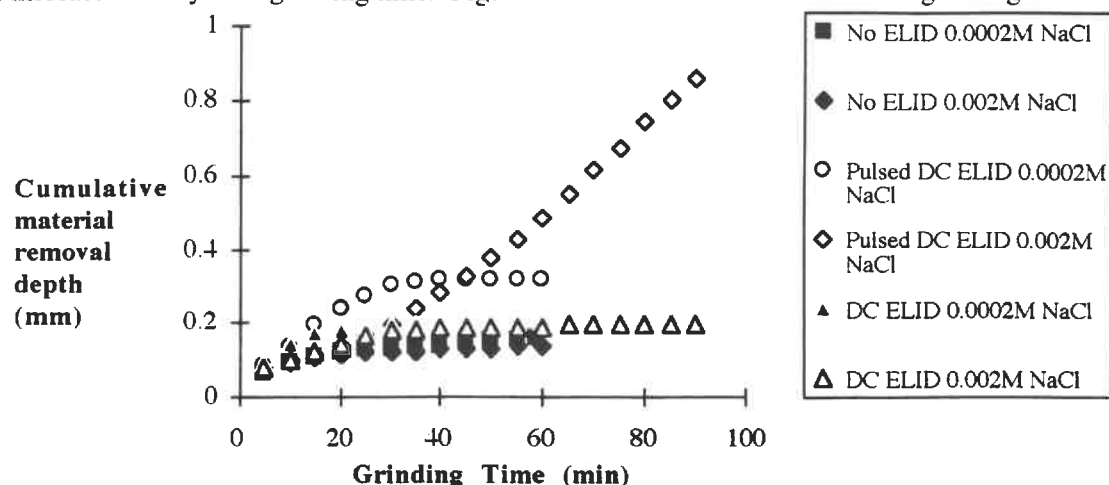


Figure 4: Cumulative depth of removal as a function of grinding time in constant-force plunge grinding with various ELID conditions. The only condition resulting in satisfactory dressing was high-current, pulsed DC ELID.

In all of the experiments involving no ELID or DC ELID, the grinding wheel became dull over a period of approximately 20 minutes, after which minimal amounts of material were removed from the workpiece. With low current pulsed DC ELID (0.0002M NaCl electrolyte), the cessation of cutting was delayed by approximately 10 minutes. With high current pulsed DC ELID (0.002M NaCl electrolyte), the depth of material removed increased linearly with grinding time, indicating successful dressing.

A practical implementation of ELID was demonstrated and studied using a diamond grit, bronze-bonded cut-off saw designed for precision, low deformation sawing and wafering³ [6]. This type of machine is often used for sample preparation. When cutting ceramics or glasses, the diamonds in the wafering blade become dull relatively quickly. Dressing is accomplished by periodic application of a porous ceramic "dressing stick" pressed against the wheel's perimeter. The potential for ELID in this application is to allow automation of the sample preparation process and to allow more uniform sawing.

An experiment was conducted to study the effect of turning on ELID when the wafering blade performance had deteriorated. Figure 5 shows the cutting rate as a function of time for a glass workpiece having a 6.4 mm x 6.4 mm cross-section. The blade was initially dressed to sharp condition and then the sample was cut 29 consecutive times. The blade started to become dull after about 10 cuts and its performance deteriorated rapidly after about 20 cuts. By the 26th cut, cutting rate had dropped by more than an order of magnitude. ELID was turned on at this point. The cutting rate returned to its original value within three cuts. Continuous ELID on a cut-off saw was also implemented successfully.

DISCUSSION

While the data in Figures 4 and 5 show that it is sometimes possible to obtain satisfactory dressing using electrolysis with bronze-bonded wheels, they also show that such dressing demands appropriate electrochemical conditions. Further, it is apparent that the appropriate ELID conditions are affected by grinding geometry, electrode size, inter-electrode gap, electrolysis current, electrolyte chemistry, and workpiece material properties. Since the purpose of dressing is to remove dull abrasive grains, one can also expect the appropriate ELID conditions to depend upon the rate of wear of the abrasive edges.

²In plunge grinding, the cumulative removal volume is the wheel width multiplied by the cumulative depth-of-removal or infeed.

³Buehler ISOMET Low Speed Saw

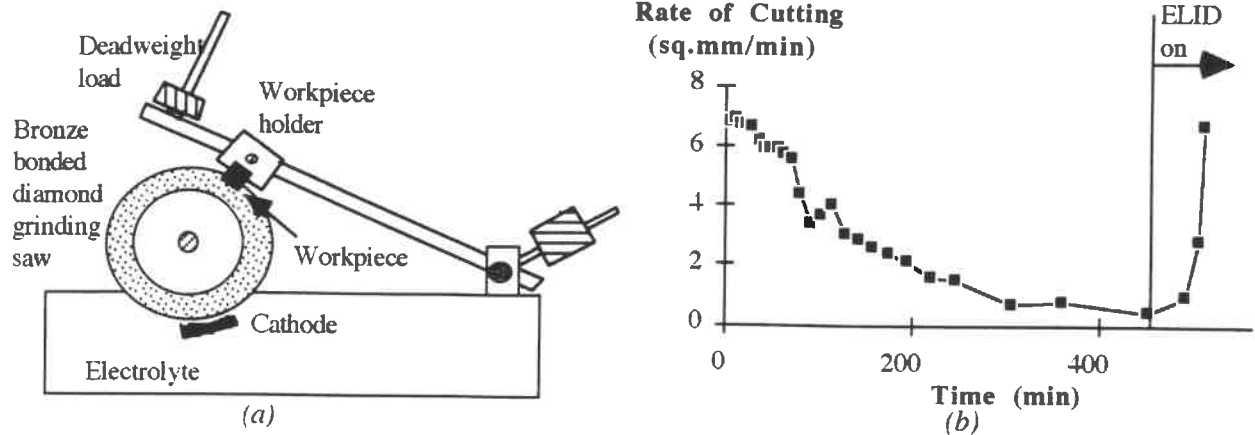


Figure 5: Cut-off Saw Wafering, (a) Buehler ISOMET Low Speed Saw retrofitted with ELID. (b) Effect of turning on ELID once the blade is dull.

One deterministic way to approach selection of ELID conditions would be to evaluate three parameters of the grinding system; for given grinding and cathode geometry.

- Evaluate the rate of growth of the metal oxide film (V_G) without grinding contact.
- Evaluate the rate of wear of the oxide film (V_W) with grinding contact (no ELID).
- Evaluate the useful life (t_D) of the wheel subject to grinding without ELID.

For successful ELID during actual grinding, $V_W \geq V_G \geq D/t_D$, where D is the diamond abrasive size. That is, the rate of oxide film growth must be equalled or exceeded by the rate of film erosion as a result of contact between the wheel and the workpiece, to ensure that the film does not envelop the grinding abrasive. Furthermore, the rate of removal of bond ($\sim V_W$) must be sufficient to cause a diamond embedded to a depth D to fall out of the grinding wheel before it becomes too dull.

A value for V_G can be obtained from the slope of a graph similar to Figure 2. This slope is strongly dependent on the electrode geometry and it will scale approximately linearly with electrolysis current for a given geometry. A value of t_D can be obtained from a graph, similar to the one in Figure 1.

We have not yet made a measurement of V_W , but for the ceramic workpieces that we have used to date, the oxide film always seems to be eroded by workpiece contact at a sufficient rate so as to avoid unchecked film growth. Using an ultraprecision grinding apparatus, we are currently attempting to measure V_G , t_D , and V_W for a particular combination of wheel, electrode geometry, electrolyte, and workpiece to deterministically implement ELID in a microgrinding operation.

REFERENCES

- [1] Bifano, T.G., Yi, Y. and Kahl, K., "Fixed Abrasive Grinding of CVD SiC Mirrors," *J. Precision Engineering*, April 1993, 16(2), pp. 109-116.
- [2] H. Ohmori, T. Nakagawa. Institute of Industrial Science. University of Tokyo. "Mirror Surface Grinding of Silicon Wafers with Electrolytic In-Process Dressing." *Ann. CIRP*. 39/1. 1990. p. 329-332.
- [3] H. Ohmori. "Efficient and Precision Grinding Technique for Ultraprecision Mirror Surface Machining." *Japan Soc. Prec. Eng.* Vol. 26. No. 4. Dec. 1992. p. 273-278.
- [4] H. Ohmori. "Efficient and Precision Grinding Technique for Ceramics with Electrolytic In-process Dressing (ELID)." *NIST Special Publication 847*. June 1993. p. 359-366.
- [5] E. Welch. "Electrochemical Dressing of Bronze Bonded Diamond Grinding Wheels." Master's Thesis. Boston University. 1993.
- [6] R. Krishnamoorthy. "Electrolytic In-Process Dressing of Bronze Bonded Diamond Wheels." Master's Thesis. Boston University. 1994.

ACKNOWLEDGMENTS

Experiments on the Ultraprecision Grinding Apparatus were performed by Boston University undergraduates Amanda Kabak and Tony DaSilva as well as Florida Institute of Technology undergraduate Katresa Banks. Funding was provided by the National Science Foundation [DMI9202377] and wheels were donated by Norton Company.

IN-PROCESS DRESSING OF GRINDING WHEELS FOR BRITTLE MATERIALS

Keith Sharp, R. O. Scattergood

Precision Engineering Center, North Carolina State University

INTRODUCTION

Hard, brittle materials such as ceramics or glass are typically fabricated using grinding wheels containing a large number of diamond grits in a plastic or metal binder. Workpiece material is removed by interaction with the diamond grits which also creates debris and dulls the grits. In many cases the grinding force, particularly the normal component, grows as the wheel becomes "worn" and can cause the grinding to transition from the ductile to the brittle regime [1]. The force increase can occur due to degradation of the diamond grit, grinding debris hiding the grits, or indenting/wearing the grits to below the level of the binder.

Dressing the wheel will usually return the forces to their original levels [2] presumably because it removes debris and some of the binding agent to expose new diamonds. This can be performed mechanically using abrasives sticks of fiber filled wipers [3], or chemically using electrolysis. Electrolytic In-process Dressing (ELID) uses electrochemistry to dress the wheel. The work presented was designed to investigate ELID parameters and their relative effect on the ductile regime grinding process for brittle materials. All of the force data presented involves ductile grinding.

DESCRIPTION OF EXPERIMENTS

Each test consisted of a series of grinding infeeds. During each infeed a piezoelectric translator (PZT), controlled with a closed loop feedback system, plunged the workpiece into a cup grinding wheel at the desired rate to a depth of 14.5 μm and then retracted to zero. The workpiece was then advanced to the previous 'hold' position with an 80 pitch screw and the next infeed started. Normal force, thrust force, and position were recorded continuously by a PC based data acquisition system while ELID amperage and power supply voltage were also recorded.

Plunge grind tests for similar conditions were conducted with and without ELID to determine the effect of ELID on the grinding forces and the workpiece surface features. Additional tests were conducted to determine the influence of the ELID parameters on the process. To determine the combined effect of ELID and grinding parameters, linear statistical modeling was used to select ten test conditions to evaluate the effect of wheel diamond concentration, workpiece material, infeed rate, ELID voltage and pulsed DC frequency on the process. High and low values of each parameter were selected as were mid values in a pattern that would allow the determination of the importance of that parameter. The ten test conditions selected for the tests are shown in Table 1.

Test No.	Diamond Conc(%)	ELID voltage (V)	Workpiece Material	Infeed Rate (nm/rev)	ELID Frequency (μsec on/off)
1	50	30	Bk7	28	9
2	50	30	Sapphire	28	4
3	50	90	Bk7	4	9
4	50	90	Sapphire	4	4
5	75	30	Bk7	4	4
6	75	30	Sapphire	4	9
7	75	90	Bk7	28	4
8	75	90	Sapphire	28	9
9	50	60	Fused Silica	14	7
10	75	60	Fused Silica	14	7

Table 1 : Test Parameters for Experimental Sequence

EXPERIMENTAL RESULTS

The sequence of experiments described in Table 1 created a large body of data to be used to determine the influence of each variable. The hypothesis was that the grinding forces are an important measure of the process. Changes in these forces, particularly the normal force, can have a significant effect on the damage to the workpiece surface [3]. Therefore measures of the force necessary to remove the material was deemed to be key parameters for comparison.

Electrolytic In-Process Dressing (ELID) had a significant effect on the normal forces generated during ductile regime grinding. Figure 1 shows a comparison of the normal grinding force observed during grinding of BK-7 glass specimens conducted with and without ELID using a 75% concentration iron-bonded wheel, 50 μm diamond grit, and 28 nm/rev infeed. Without ELID, the normal force continues to grow over the entire experiment while the addition of ELID makes the force become asymptotic to a value less than a third of the maximum force measured without ELID.

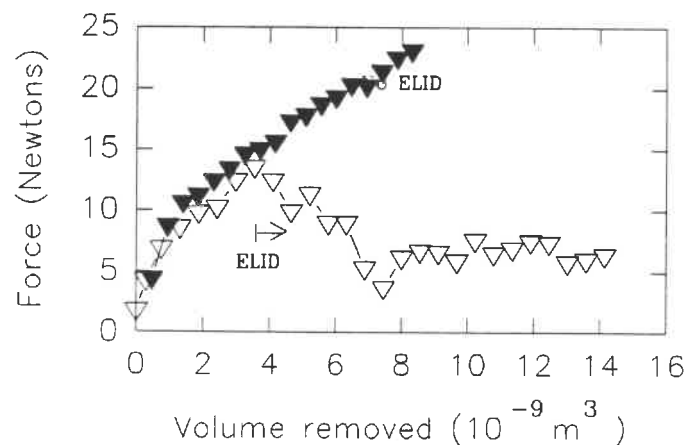


Figure 1: Comparison of Normal Grinding Force with and without ELID (BK-7 Glass at 28 nm/rev infeed)

Figure 2 shows the mean forces (average force per revolution of the wheel) versus volume removed for the 10 test conditions described in Table 1. All of these experiments used ELID but because the test conditions are randomized, it is difficult to see trends solely by looking at the force curves. Since the ELID parameters for a given test condition may not be optimum the forces do not all become constant after a short time as in Figure 1.

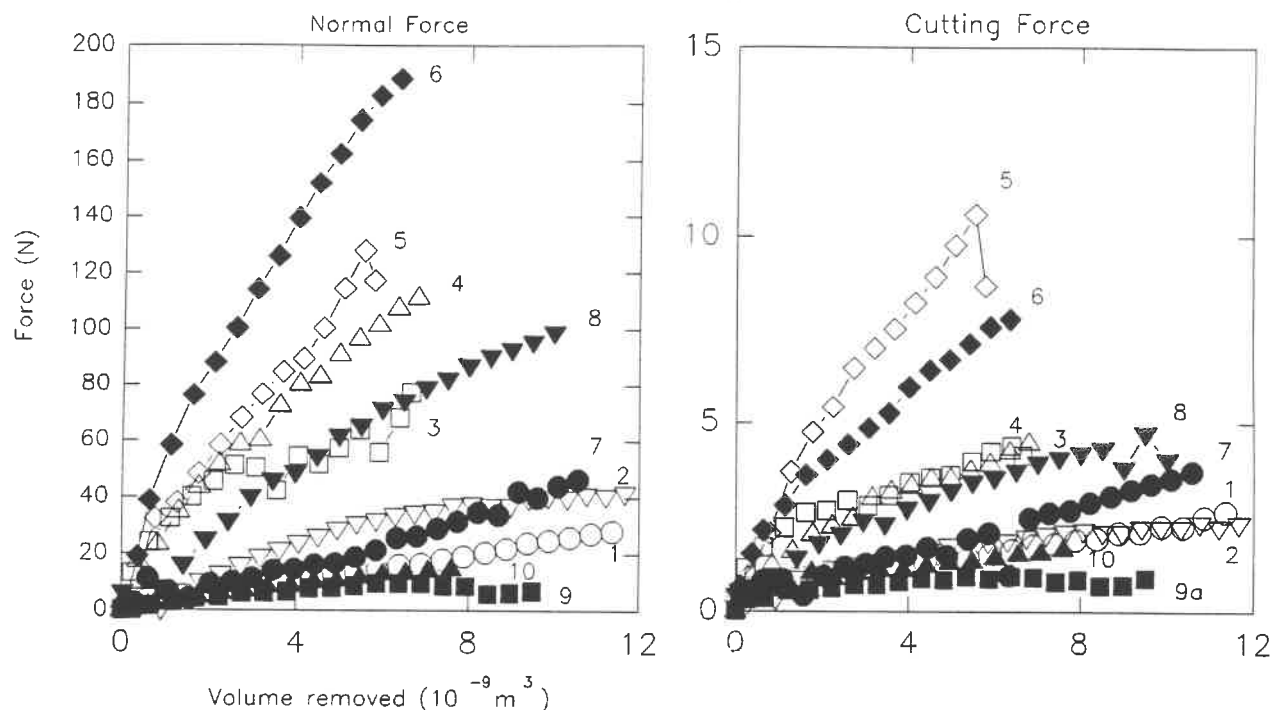
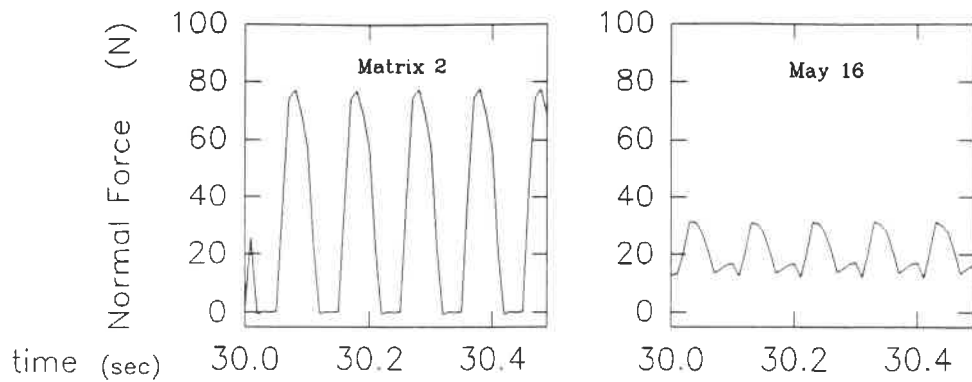


Figure 2: Normal and Cutting Force Data as a function of Volume removed for the 10 Test Conditions of Table 1. (Force indicated is mean force per wheel revolution)

The grinding force from each test is assumed to be a linear combination of the various test parameters. Matrix solutions were sought that solve for the best-fit coefficients indicating the relative importance of each variables. Since the statistical comparison must be made at a specific volume removed, the volume near the end of Test 5 ($5.7 \times 10^{-9} \text{ m}^3$) was chosen to compare the test results. The experimental coefficients were fit to the average normal force. Further comparisons include least-squares linear slopes of the force curves shown in Figure 2.

The amount of contact between the grinding wheel and the workpiece affects the peak grinding forces and these forces will vary significantly with the runout of the grinding wheel. Figure 3 shows a half-second of normal force data for two representative test runs; one has high runout and high peak forces with a loss of contact; the other has one-third of the runout and lower peak forces with continuous contact. Therefore, mean, rather than peak force seems to be more meaningful since the work done to remove a given volume of the workpiece should be invariant in steady state grinding with the same wheel, workpiece and infeed. For this reason the mean force is shown in Figure 2 and is used to compare the test results.



(a) Runout, 10.5 μm

(b) Runout 3.3 μm

Figure 3: Normal force data for several wheel rotations
(Sapphire at 28 nm/rev infeed)

One ambiguity in the test results was the low forces measured for the Fused Silica tests. Fused Silica has a hardness similar to BK7 (8 GPa for Fused Silica and 9 GPa for BK7) and much less than sapphire (27 GPa). Since hardness predicts the force required to push a grit into a workpiece surface, Fused Silica and BK7 forces should be similar, unless additional reactions takes place.

If the Fused Silica tests are considered anomalous, several parameters appear statistically significant. The normal force increased with hardness and pulsed DC cycle time, and decreased with infeed, diamond concentration, and ELID voltage. The slope of the normal force to volume removed increased with hardness, and decreased with infeed, diamond concentration and ELID voltage. The cutting force decreased significantly with infeed and the slope of the cutting force to volume removed decreased with both infeed and concentration.

CONCLUSIONS

In the direct comparison grinding tests, ELID reduced the grinding forces. The statistical analysis also shows that the normal force depends on the ELID parameters, although the cutting force show little such dependence. The normal force increase with increasing ELID voltage may reflect a decrease in surface damage of the workpiece surface. Any amount of surface fracture reduces the energy required to remove a unit volume of material, which in turn decreases the grinding forces. Further analysis is being performed to include the effects of some surface damage and how that could relate to the change in the normal forces.

REFERENCES

- [1] Dow, T. A. and Scattergood, R.O., Jap. Soc. for Prec. Eng., V. 56, No. 5, May 1990.
- [2] Ohmori, H., Mach. of Adv. Mat., Proc. of Int. Conf. on Mach. of Adv. Mat., July 1993.
- [3] Kohzo A., and Yoasunaga ., Proc ASPE, V.6, Oct 1992
- [4] Tanikella, B.V. and Scattergood, R.O, Jour. Amer. Ceramic Soc., 78 [6] , 1995.

Spring-Dominated Regime Design of a High Load Capacity, Electromagnetically Driven X-Y-θ Stage

Kuo-Shen Chen¹, Andre Monterio², David Trumper¹, Stuart Smith³, and Mark Williams¹

We have designed and constructed a three degree of freedom electromagnetically-actuated flexural stage as an alternative approach to existing piezoelectrically driven fine-positioning stages. As shown in Figure 1, our stage design consists of four vertical steel flexures arranged like the legs of a table, supporting a platform that serves as both a sample holder and a kinematic mount for the position sensor surfaces. The steel flexures provide high system stiffness and rejection of external vibration and noise. A thin film of high viscosity silicone fluid (GE Viscosil) is added between the stage bottom surface and the base to provide system damping. Six electromagnet actuators are used to control the three degrees of freedom (x , y , θ_z) not restrained by the flexures. The x and y degrees of freedom are the two major axes of motion. The θ_z degree of freedom is also controlled since this axis is not restrained by the flexure system. The sample holder also serves as the position sensing target. A kinematic mount is used to separate the force and metrology loops so that distortions in the stage do not couple into the position feedback.

The basic working principle of the stage is shown in Figure 3. The magnetic actuators exert an attractive force and are arranged in opposing pairs. The stage legs provide spring forces and the viscous layer provides damping to help stabilize the system. The stage system thus can be modelled as a spring-mass-damper system driven in three degrees of freedom by six electromagnets of the type shown in Figure 4. The system dynamics in each degree of freedom are therefore

$$m\ddot{x} + b\dot{x} + kx = C \frac{i^2}{(x + g_0)^2} \quad (1)$$

where C is the force constant, and x is the air gap. Since the actuators are highly nonlinear, they are linearized by the control computer in real time given the desired force and the actuator gap. The relationship between current, force, and air gap is determined using a fixture developed previously [PHI94]. From this data, we measure a value of $C = 1.3 \times 10^{-5} \text{ Nm}^2/\text{Amp}^2$ and $g_0 = 30 \mu\text{m}$.

Dynamic testing was used to obtain the system parameters. The major translation modes exhibit a 125 Hz natural frequency and a damping ratio of 1.2. The minor rotational mode has a 210 Hz natural frequency and a damping ratio of 0.6.

The overall system operation is shown in Figure 2 and Figure 5. Three capacitive position sensors are used to measure the stage position. Their measurement range is 120 μm with a 5 nm resolution. The sensor gain is 200 mv/ μm . The 12 bit A/D converter gets the position information into the host computer. The controller implements a nonlinear

¹ Laboratory for Manufacturing and Productivity, Massachusetts Institute of Technology.

² Center for Nanotechnology and Microengineering, University of Warwick, United Kingdom.

³ Mechanical Engineering Department, University of North Carolina at Charlotte.

feedback compensation algorithm. It consists of two feedback loops: an inner loop which utilizes feedback linearization to transform the nonlinear plant into a form that is linear and operating point invariant, and an outer loop which uses a standard linear controller to stabilize the system. A digital proportional plus integral (PI) control technique is applied in this outer loop. The control efforts are output to amplifiers via D/A converters. Finally the currents from the power amplifiers produce desired control forces in the six magnetic bearings to control the motion of stage.

We use the C language to implement the digital control law on a Pentium 5 90MHz computer to control the motion of the stage. The system shows a position noise resolution of two least counts of the analog-digital converter that is used to read capacitive probe output. This translates to a peak-to-peak position noise of about 50 nm. The experimental data show that the system resolution is dominated by the quantization of the analog to digital converters. The present system bandwidth is 90 Hz. Figure 6 shows the stage control performance under several proportional and integral gains. The power consumption is less than 1 watt under normal operation.

Finally, the stage has been integrated with a stylus to allow surface roughness scanning. The system set-up is shown in Figure 7. The sample moves with the stage while the stylus is stationary. Figure 8 is a typical surface image taken using our stage to provide the scanning motions. The maximum scanning area is $100\text{ }\mu\text{m} \times 100\text{ }\mu\text{m}$.

In summary, we have developed and experimentally demonstrated an electromagnetically-driven stage with large motion range, good dynamic response, good base vibration rejection, and reasonable system bandwidth. Some potential applications of this stage include scanning tunneling microscopy, surface profilometers, and as the fine positioner for a coarse/fine positioning system.

Reference:

- [Che95] **Chen, K. S. (1995):** *"A Spring-Dominated Regime Design of a High Load Capacity, Electromagnetically Driven X-Y- θ stage"*. Master Thesis, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA 1995.
- [PHT94] **Poovey, T., Holmes, M., and Trumper, D. (1994):** "A kinematically Coupled Magnetic Bearing Calibration Fixture", *Precision Engineering*, Vol 16, No. 2, Apr., 1994.
- [Slo92] **Slocum, A. H. (1992):** *Precision Machine Design*, Prentice-Hall Inc., Englewood Cliffs, NJ, 1992.
- [SmC91] **Smith, S. T., and Chetwynd, D. G. (1991):** *Foundations of Ultraprecision Mechanism Design*, Gordon and Breach Science Publishers, U. K., 1991.
- [Tru90] **Trumper, D. L., (1990):** *"Magnetic Suspension Techniques for Precision Motion Control,"* Ph. D. Thesis, M. I. T., Cambridge, Massachusetts, 1990.
- [WiT93] **Williams, M. E. and Trumper, D. L. (1993):** "Materials for Efficient High-Flux Magnetic Bearing Actuators", 2nd International Symposium on Magnetic Suspension Technology, Seattle, WA 1993.
- [WoM68] **Woodson, H. H., and Melcher, J. R. (1968):** *Electromechanical Dynamics-Part I, II*, John Wiley & Sons Co., 1968.

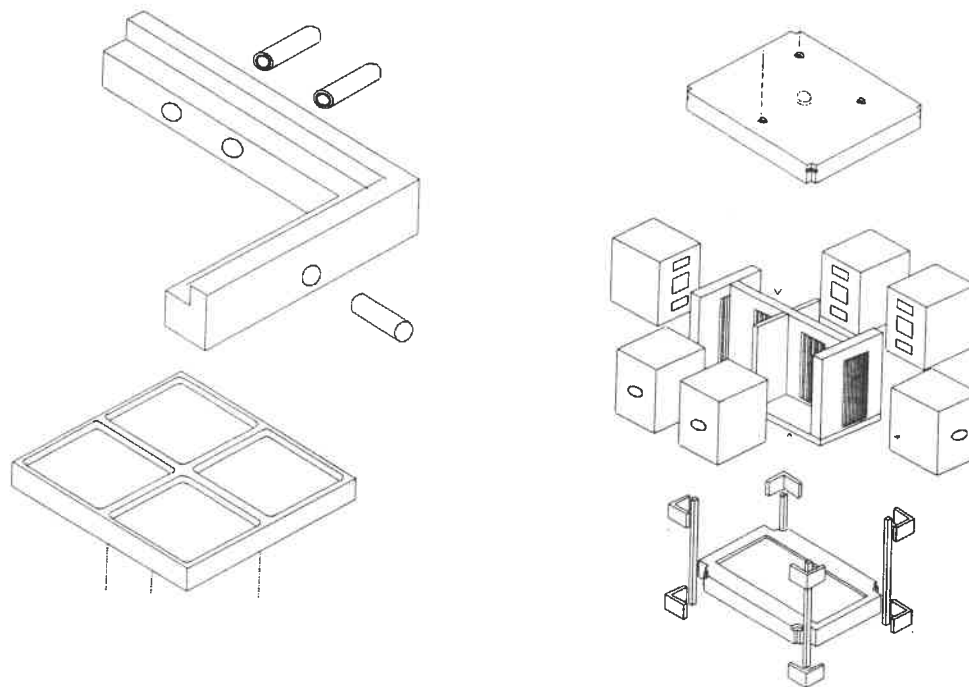


Figure 1. Exploded view of stage

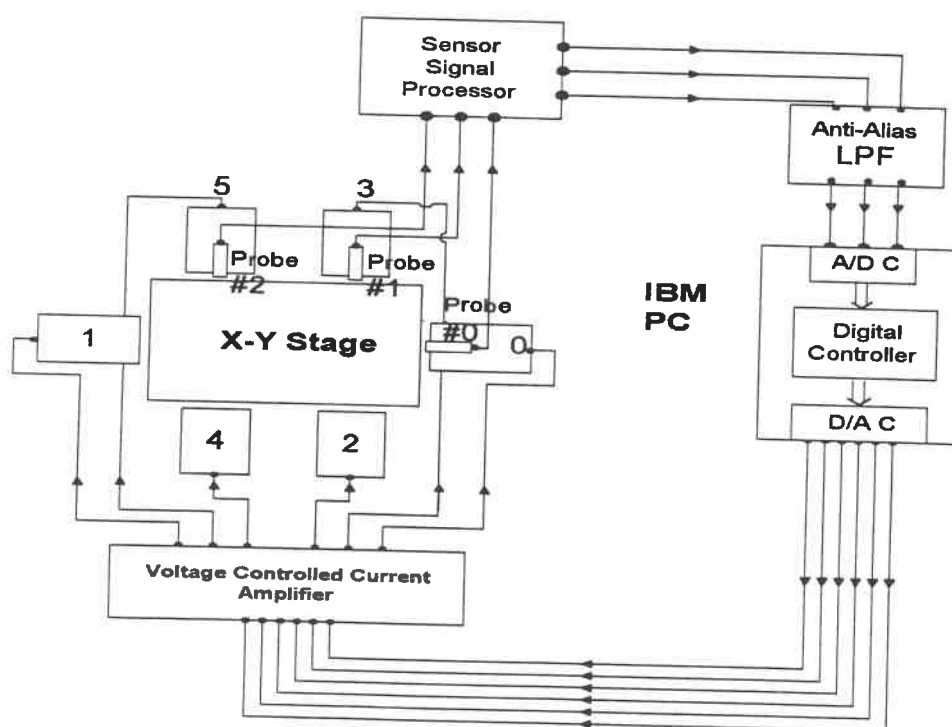


Figure 2. System block diagram

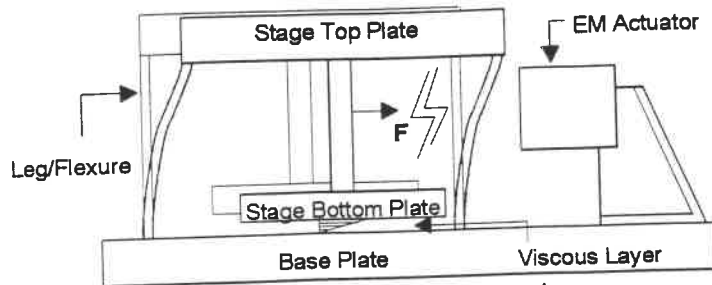


Figure 3. Schematic view of stage motion

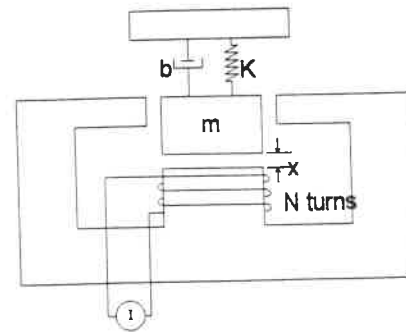


Figure 4. System modelling

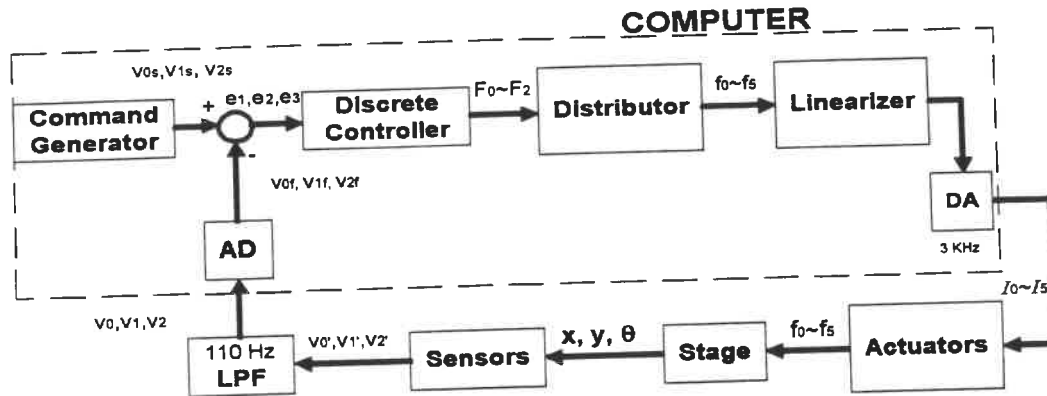


Figure 5. Control functional diagram

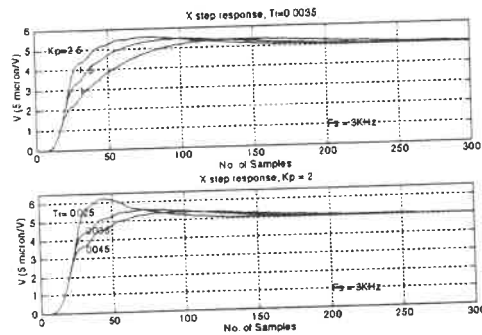


Figure 6. Step Responses

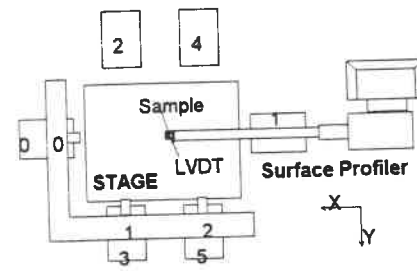


Figure 7. Surface scanner setup

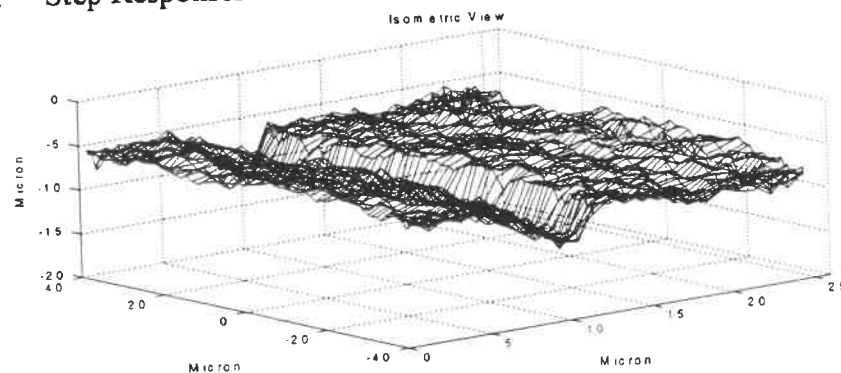


Figure 8. Surface scanning result

A VERSATILE XY STAGE WITH A FLEXURAL SIX-DEGREE-OF-FREEDOM FINE POSITIONER

Julie A. Gargas, Rick K. Dorval, David Mansur, Hy D. Tran,
David Carlson, and Michael Hercher
OPTRA, Inc.

461 Boston Street, Topsfield, MA 01983

Semiconductor microlithography processes require large ranges of motions in X and Y (200 mm or more), combined with high positioning accuracy in all six degrees of freedom relative to a reference frame defined either by the image-forming optics, or by the beam apparatus. The dynamic ranges for Z motions, and for θ_x , θ_y , and θ_z rotations, depend on the precision of the mechanical ways and the flatness of the substrate: ranges of $\pm 20 \mu\text{m}$ for Z and $\pm 0.1 \text{ mrad}$ for rotations are typical. A six-degree-of-freedom short-stroke flexural stage, having a $40 \mu\text{m}$ range of motion, can be used to correct for errors in the motion of an air-bearing-based XY stage. Scale-based metrology can both reduce Abbe errors, and very nearly eliminate the turbulence errors (i.e. errors due to localized fluctuations in air temperature or pressure) associated with laser interferometers. A novel XY planar encoder scale is used to measure X and Y with a resolution of 0.3 nm, and to monitor yaw with a precision of $\pm 60 \text{ nrad}$. Preliminary work has demonstrated 50 mm moves at speeds up to 400 mm/s, and fine motions with a repeatability of $\pm 5 \text{ nm}$. The overall system design is presented, including discussions of the flexural fine positioning stage, the XY planar encoder design, and the architecture for the overall control system.

INTRODUCTION. Critical Dimension (CD) and overlay requirements in semiconductor lithography impose stringent requirements on stage technology. Overlay requirements are among the most difficult technical challenges, requiring advances in stage and alignment technologies, environmental controls, and metrology¹. Overlay requirements are typically 30% of the lithography resolution. As processes move from $0.35 \mu\text{m}$ to $0.25 \mu\text{m}$, to $0.18 \mu\text{m}$, and to smaller geometries, overlay metrology requirements will move from 10 nm to 7.5 nm to 5 nm.

High precision X-Y stages currently use 2-frequency laser interferometers for metrology. One stage is stacked on the other, and the top stage carrying the workpiece (i.e. the wafer) also carries an "L" shaped mirror². The accuracy of laser interferometers is subject to errors due to variations in index of refraction in air due to temperature, pressure, humidity, and turbulence³. The use of a scale-based metrology can reduce the sensitivity of the metrology to atmospheric variations⁴. While plane-mirror interferometers can be used in an X-Y application, it is difficult to minimize Abbe errors in an X-Y stage using one-dimensional scales⁵. Therefore, a two-dimensional (planar, or X-Y) scale-based metrology system has been developed⁶. The use of a scale-based metrology can nearly eliminate turbulence errors associated with laser interferometers. In addition, the placement of a scale system directly underneath the functional point (imaging optics in a lithography system) can minimize Abbe offset and cosine errors. Finally, the optics used in a scale-based system offer a cost advantage over a laser interferometer based system.

SYSTEM OVERVIEW. In order to use the planar encoder effectively, it is necessary to design the stage system with the metrology in mind. To minimize Abbe offset and alignment errors, the metrology should be placed in line with the functional point, i.e. directly underneath the workpiece⁷. The X-Y stage system is based on a stacked stage concept, but designed as an "H" frame, with one stage carried inside the other stage. This is useful, not only in reducing the overall height of the system, but also for minimizing the distance between the scale/readout and the workpiece. To correct for error motions in the slideways, a short stroke flexural stage is used for fine positioning. Figure 1 shows the system design.

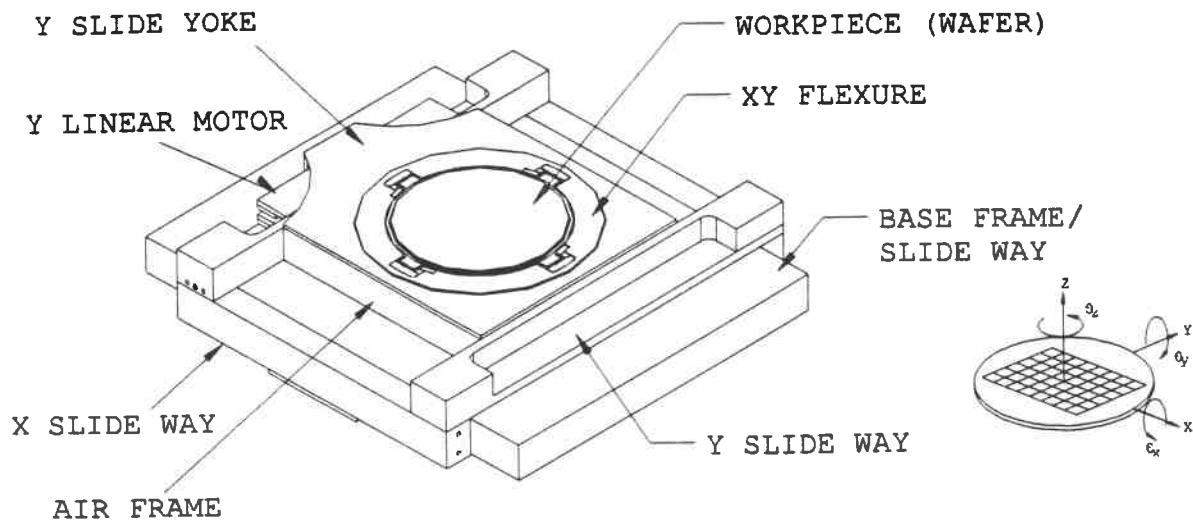


Figure 1. Overview of X-Y stage system using an X-Y scale as the basis of the metrology.

The stage is based on air bearings driven by linear brushless DC motors. We are using a 150 mm by 150 mm scale grid as the metrology, with a resolution of 0.3 nm. The grid is mounted on the back of the wafer carrier, which is supported by a 6 degree-of-freedom piezoelectrically driven flexural stage. This short stroke stage is attached to the air bearing slideways. The short stroke flexure stage has a 40 μm range of motion in X and Y; a 20 μm range of motion in Z; and rotational range of motion of 0.1 mrad. This is sufficient to compensate for error motions in the air bearing stage, which moves over a 150 mm by 150 mm workspace.

METROLOGY SYSTEM. The XY planar encoder metrology system consists of a crossed scale grid, or coarse grating whose area covers the desired ranges of measurement in the X and Y directions. The spacing of the rulings is 10 μm ; this spatial frequency is low enough to allow photographic replication of the gratings and high enough to allow for sub-nanometer interpolation. The readout heads interferometrically sense the relative displacement of the XY encoder such that there are two cycles of phase change for a displacement equal to one ruling spacing. Two readout heads are used to measure X, Y, and to derive Θ_z . Figure 2 shows the layout of the planar encoder system.

The operation and measurement capabilities of the XY planar encoder readout can be fully understood in terms of the operation of a 1-dimensional linear encoder, e.g. the OPTRA Nanoscale. The linear encoder is illuminated by a beam from a laser diode at normal incidence, and the reflected diffracted light is spatially filtered to select just the ± 1 diffracted orders. These two beams are brought together again by an optical system. In the region where the two beams intersect, a fringe pattern is formed. The magnification of the optical system is designed so the spacing between the interference fringes is 45 μm . A special detector array is located in the plane of the interference fringe pattern. This detector array consists of 90 linear elements on 15 μm centers, with every third element in parallel. When the encoder moves relative to the readout head, the fringes move relative to the detector, and sinusoidal signals are generated at each of the three detector outputs. It should be noted that the metric for the OPTRA Nanoscale is the one-half pitch of the grating; *not* the wavelength of the illuminating laser. The three detector signals, R, S, and T are 120° apart in phase.

In both Michelson interferometers and interferometric readout sensors for linear encoders, the interferometric phase is proportional to the mechanical displacement. The usual relationship (for a 2-beam interferometer) between observed intensity, I , and interferometric phase, ϕ , is:

$$I = I_1 + I_2 \cos(\phi),$$

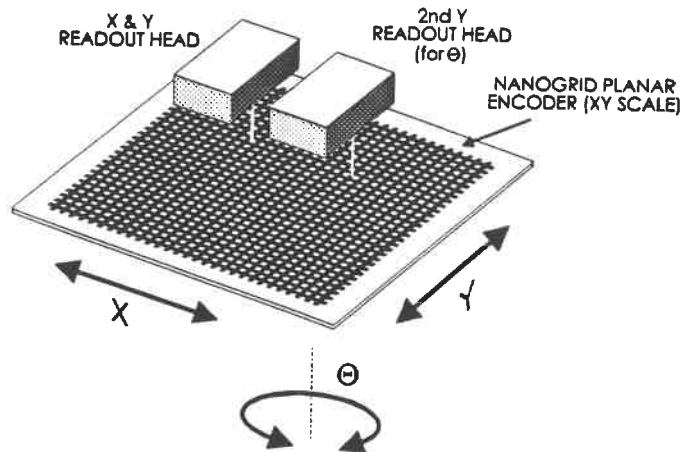


Figure 2. Physical layout of an XY planar encoder.

where I_1 is the DC light level and I_2 is the intensity amplitude of the phase-dependent part of the signal. In the Nanoscale technique, three separate interferometric intensity measurements are made which differ only in their interferometric phases (in increments of $2\pi/3$ or 120°):

$$R = I_1 + I_2 \cos(\phi),$$

$$S = I_1 + I_2 \cos(\phi + 2\pi/3),$$

$$T = I_1 + I_2 \cos(\phi + 4\pi/3),$$

These three equations can be solved for ϕ so that the measurement is independent of temporal fluctuations in I_1 and I_2 regardless of their frequencies.

Measurement of Z motion, along with pitch and roll of the stage, is done using capacitive or other proximity sensors.

FLEXURAL SHORT STROKE STAGE DESIGN. The flexural short stroke stage design is shown in Figure 4. The short stroke stage consists of a number of independent modules. There is a Θ_z flexure, which is attached to an X-Y flexure, which is attached to a Z, Θ_x , and Θ_y flexure. The X, Y, and Θ_z motions are driven by independent piezoelectric actuators. The Z, Θ_x , and Θ_y motions are driven with three coupled piezoelectric actuators: Driving these three actuators together gives Z motion; while driving them differentially yields tilt motions.

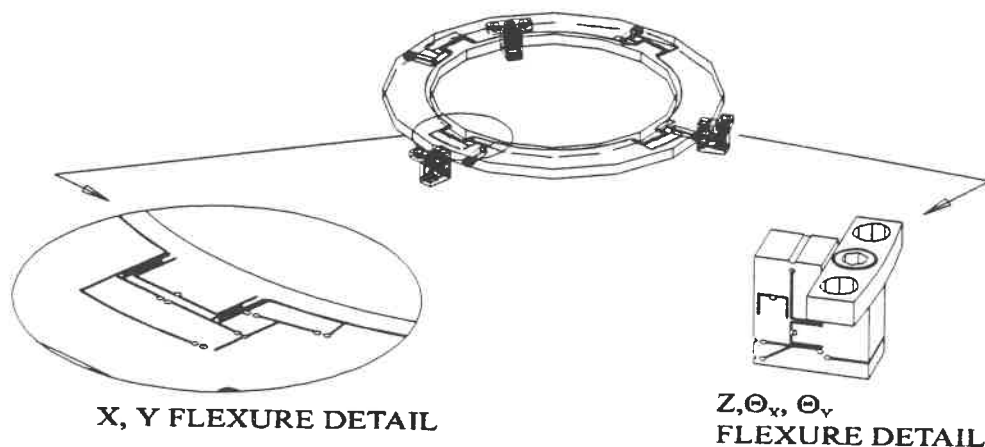


Figure 3. Short stroke flexural stage is designed as an assembly of independent modules. The scale and readout head is mounted underneath the flexural stage; the workpiece on top of the flexural stage, minimizing Abbe offsets and alignment errors.

CONTROL STRATEGY. Coarse/fine systems have multiple actuators on an axis, and require extra care in servo implementation. Since the X and Y axes are orthogonal, they are reasonably decoupled. Design and analysis can be done on a single axis (say, X direction motion). Modeling the dynamics using state-space methods help tune the servo more effectively, while actual control mechanization is via successive loop closure, using commercial motion control boards. Figure 4 shows preliminary performance of the system.

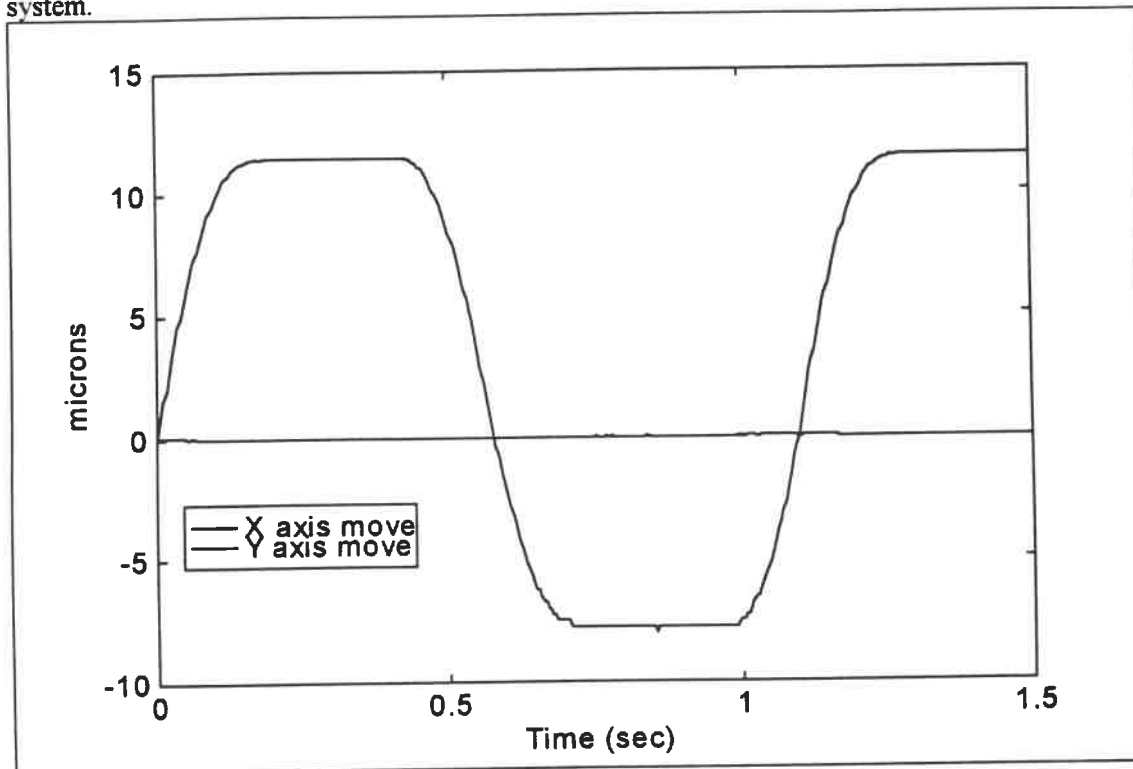


Figure 4. Control of the flexure shows no cross-coupling of X and Y and position stability of ± 5 nm.

ACKNOWLEDGMENTS. This work was funded, in part, through the support of ARPA contract DAAH01-94-C-R271 and ONR contract N00014-94-C0033.

REFERENCES

- ¹ *The National Technology Roadmap for Semiconductors*, Semiconductor Industry Association, 1994.
- ² A. Slocum, *Precision Machine Design*, Prentice-Hall, 1992.
- ³ W. Tyler Estler, "High-accuracy displacement interferometry in air," *Applied Optics* 24 (6), pg. 808-815, 1985.
- ⁴ G. Wyntjes, D. Mitchell, and M. Hercher, "Scales vs. Interferometers: A Comparison of Metrology Systems," *Proc. 6th ASPE*, Santa Fe, 1991.
- ⁵ H. Kunzmann, T. Pfeifer, and J. Flugge, "Scales vs. Laser Interferometers: Performance and Comparison of two Measuring Systems," in *CIRP Annals* v.42/2, pg. 753-767, 1993.
- ⁶ G. Wyntjes and M. Hercher, "XY Positioning Stages: A New Approach," in *Proc. 6th IPES*, Braunschweig, 1991.
- ⁷ J. Bryan, "The Abbe Principle Revisited: An Updated Interpretation," *Prec. Eng.* 1, pg. 129-132, 1979.

Long Range, Direct Drive XY-($\Phi=0$)-Stage with Sub-micrometer Resolution

G.M. Bonnema, F. Auer, J. de Lange, and H.F. van Beek
Delft University of Technology, Laboratory for Micro Engineering, The Netherlands

1 Introduction

In precision engineering, often an accurate movement in one plane is needed. Processes where this is the case are: laser cutting, coordinate measuring, photo plotting, and step-and-repeat processes. There is a large variety of stages available that perform this XY-movement. Most of these stages use the stacked structure, where the X-drive carries the entire Y-drive, or vice versa.

At the Laboratory for Micro Engineering at the Delft University of Technology, it was recognized that a *direct drive* system can have a better dynamic performance (also see [1]). In such a system, the drives are not stacked, but all work between the rigid world and the moving part.

In this paper, an XY-stage is presented, based on this direct drive principle. Using linear motors and feedback control, a position stability better than $0.2\ \mu\text{m}$ is achieved. The angle of rotation around the Z-axis, Φ , is electronically controlled to zero within $0.36\ \mu\text{rad}$. The propulsion system allows a travel of $100 \times 100\ \text{mm}^2$.

This research was sponsored by Philips CFT of the Nederlandse Philips Bedrijven B.V.

2 Design Approach

In this project the *Mechatronic Approach* is applied. Here, unlike in 'traditional engineering', the cooperation between different fields of engineering is taken into account from the beginning.

In order to eliminate cross-talk between the guidances of the X- and the Y-drive, only one guidance system is used. Thus, only one moving part (*rotor*) remains, that moves in the XY-plane.

With this direct drive setup, the dynamics of the system can be improved compared to the 'traditional' stacked configuration. These better dynamics are partly due to the fact that no trans-

mission is needed between motion generating element and rotor. More important is the fact that the vertical dimension of the rotor can be kept small, minimizing the out-of-plane forces. Thus, the stage will be much less subject to tilting.

It is possible to design the system such that there are not many parts that need accurate manufacturing, reducing the cost of the system. In the system presented here, the only expensive parts are the sensor system, and the electronics used to implement the controller.

3 System Description

In this Section, the functions of the system are described separately. However, according to the mechatronic approach, all have to be considered simultaneously during the design phase.

The requirements on the system are:

- Contact-free operation,
- Travel $100 \times 100\ \text{mm}^2$,
- Maximum speed $200\ \text{mm/s}$,
- Bandwidth $100\ \text{Hz}$,
- Position accuracy better than $2\ \mu\text{m}$,
- Angular accuracy better than $20\ \mu\text{rad}$, and
- Acceleration up to $20\ \text{m/s}^2$.

3.1 Propulsion

For a contact-free, direct drive XY-stage, propulsion is most practically implemented with *linear motors*. Attention has to be paid to the fact that the propulsion in one direction, does not limit the movement in the other direction.

The type of linear motor used is the *Moving Core Motor*, short: MCM (also see, [2] and [3]). This motor uses the *Lorentz-force*.

The motor consists of the following elements (see Figure 1): (1) a stationary element, generating the magnetic field, (2) a moving coil that carries the current into the magnetic field, and (3) a moving core, that carries the coil, and guides

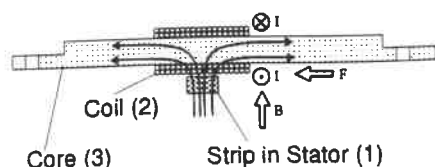


Figure 1: Essential to the Moving Core Motor is the bending of the magnetic flux along the axis of the coil. Would the field not be bent, two oppositely directed forces are generated, yielding a zero resultant force.

the flux along the axis of the coil. The MCM is simple, both in construction and use.

Essential to the MCM is the fact that the magnetic flux is bent to run parallel to the coil axis, after crossing the windings once. If the field would cross the windings at the opposite side of the coil, an oppositely directed force would be generated, yielding zero resultant force. The field is bent by a core inside the coil, made of a magnetically conducting material.

The coil, on which the Lorentz-force acts, and the core are made as one rigid unit to ensure high stiffness. Also the construction is simplified.

The system uses four motors, in pairs parallel to each other. Each pair is used for driving either the X-, or the Y- direction. Both pairs are used to generate the torque needed to control the angle Φ . The controller divides the necessary torque over the two pairs, equally.

By using a square configuration, the rotor becomes extremely simple, see Figure 2. The magnetic field is used twice: when entering and when leaving the rotor.

In the stator, also see Figure 2, the magnetic fields are generated by permanent magnets under the stator surface. The magnetic field is for every point of the travel directed towards the rotor by a steel magnetic circuit. The distance between lines of action of the forces and the center of gravity of the rotor, does not change. This reduces the complexity of the controller.

Advantages of this propulsion system are:

1. The amplifier needed for current supply is simple.
2. The rotor is easy to manufacture.
3. The rotor is a self-supporting structure. Extra elements can be connected to the rotor.

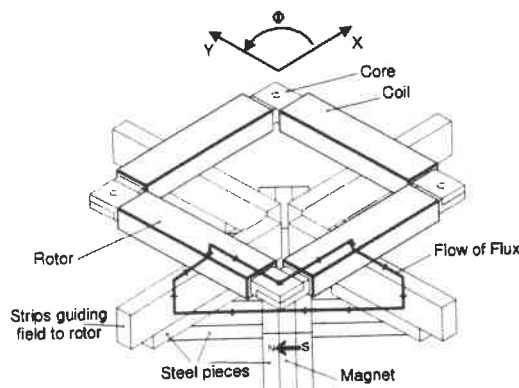


Figure 2: The propulsion system. The rotor consists of four cores in a square, each carrying a coil. The magnetic field is generated by the permanent magnet. The steel pieces and the 'strips guiding field to rotor' direct the field to the rotor. Where the field crosses the coil, the Lorentz force is generated.

Also, a high mechanical stiffness is ensured.

4. The vertical dimension of the rotor can be kept small. Thus Abbe's law is easily obeyed, and tilting of the rotor is minimized.
5. Extension of the travel of the system is expected by expanding the stator.

However, there are also disadvantages:

1. Current has to be supplied to the rotor. To minimize the parasitic forces caused by this connection, bent flexible printed circuit board is used, mechanically comparable to the buckled leaf spring.
2. The coils are used inefficiently. Only 3 % of the coil is used for force generation. Improvement is expected by only supplying current to parts of the coil.

3.2 Bearing

To ensure contact free movement of the rotor, air bearings are used. The bearing system has to reduce the number of degrees of freedom of the rotor from six to three, i.e. X, Y and Φ .

A location problem occurs when three air bearings are used. Therefore, four air bearings are chosen. Two bearings are connected via a bending lever that is halfway connected to the rotor. There, a virtual bearing point is created, suspending the rotor at three points.

The air bearings, designed in the Laboratory, are simple and function satisfactorily.

The necessary air supply hose is kept thin, made from flexible material, and bent. Thus, the parasitic forces on the rotor are small.

3.3 Position Measurement

The measurement system has to provide three coordinates: X , Y , and Φ . Because of the direct drive setup, two two-dimensional measurement systems (*Heidenhain PP109R*) are chosen. These systems use an optical grating on glass as *form standard*. The position of the measurement heads relative to the grating is obtained incrementally. Two *reference marks* define an origin, reproducible within the sensor resolution ($0.1 \mu\text{m}$).

Each PP109R system outputs two signals, S_x and S_y . The angle Φ is calculated using:

$$\Phi = K_{\text{sens}} \frac{S_{y1} - S_{y2}}{d} \quad (1)$$

where d (0.275 m) is the distance, between the two Y-measurement heads, measured parallel to the X-axis, and K_{sens} is the sensitivity of the measurement system. The second X-signal is not connected in the system presented here.

The absolute accuracy of the measurement system is $2 \mu\text{m}$, for both directions. Using equation 1, the accuracy for Φ is found to be $15 \mu\text{rad}$.

The resolution is increased with interpolation electronics (*Heidenhain EXE-units*) to $0.1 \mu\text{m}$ for the X-direction. The two Y-measurements are averaged, yielding a position resolution of $0.05 \mu\text{m}$. For Φ , the resolution is $0.18 \mu\text{rad}$.

The measurement system allows only a small angle of rotation (0.5°). An 'emergency off' is implemented in the controller that is activated when Φ exceeds a given value.

The entire electro-mechanical system is shown in Figure 3. The mass of the rotor is 2.4 kg . This includes the air bearings, the form standards and a small platform for objects. The size of the stator surface is $340 \times 340 \text{ mm}^2$. The maximum load is 0.5 kg .

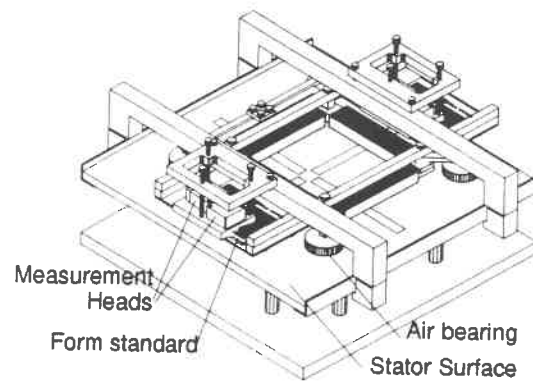


Figure 3: The entire electro-mechanical system.

3.4 Control

The complete controller is based on three independent controllers, one for each of the directions X , Y , and Φ . The design of the complete controller is simplified greatly by eliminating the dependency between Y and Φ , using coordinate transformations. One transformation is used to calculate x , y , and Φ from the signals S_x , S_{y1} , and S_{y2} . Another is used to calculate the four motor currents, I_{x1} , I_{x2} , I_{y1} , and I_{y2} from the controller outputs I_x , I_y , and I_Φ .

The controllers are of the tame PID-type. The tame PID-action is designed as to give (1) the desired bandwidth (100 Hz), (2) a stable process, (3) sufficient suppression of high-frequency noise, and (4) small enough static error ($1 \mu\text{m}$). Using the measured transfer functions of the electro-mechanical system in the three directions, the parameters for the controllers are calculated.

The controller is implemented in a *Digital Signal Processor* using a TMS320C30 processor. The software used to build the controller program is provided by dSpace GmbH.

4 System Performance

The most important performance parameters of the realized system are: top-top position stability better than $0.2 \mu\text{m}$, top-top angular position stability: $0.36 \mu\text{rad}$, stiffness larger than 610 kN/m , maximum speed: 0.21 m/s , and travel: $42 \times 42 \text{ mm}^2$.

The accuracy achieved is limited by the accuracy of the measurement system only. The top-

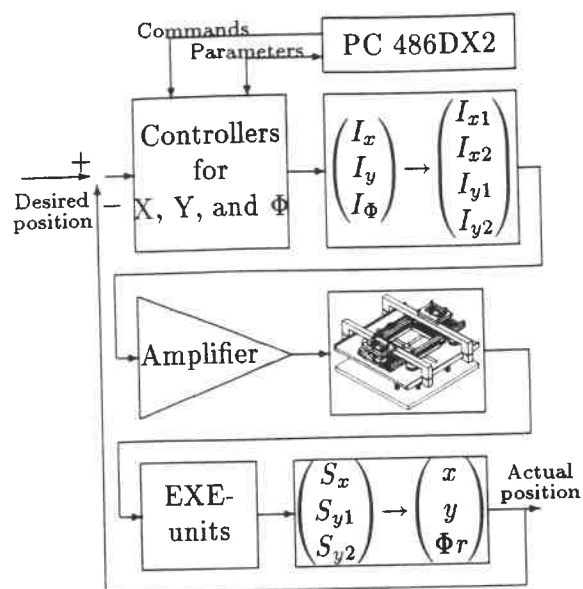


Figure 4: Setup of the controlled system. The coordinate transformations mentioned in the text are shown, as well as the interpolation of the sensor signals (EXE-units).

top position stability equals twice the measuring resolution. Therefore, higher accuracy and higher resolution are expected to be achieved with a modified measurement system.

The travel is limited by the measurement system. The propulsion system was designed for a $100 \times 100 \text{ mm}^2$ travel, which can be increased modularly by enlarging the stator.

In Figure 5 the measured response on a $100 \mu\text{m}$ step in the Y-direction is shown. The controlled system functions very well. The response is fast and sufficiently damped.

The requirement for the accelerations has not been met. The maximum acceleration measured is 1.6 m/s^2 . The forces that can be generated by the propulsion system as is, are too small to achieve 20 m/s^2 . However, it is expected that with an optimized magnetic circuit, the required forces can be generated.

5 Conclusions

A mechanically simple, contact-free XY($\Phi=0$)-stage is presented.

A new type of linear motor is presented, the

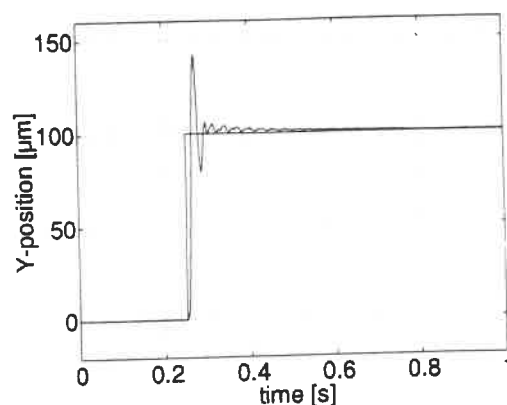


Figure 5: Measured step response of the system in Y-direction.

Moving Core Motor. This motor is easy to use and to manufacture. Moreover, the position accuracy of this type of motor is limited by the measurement system used only. This motor is suited for applications where one direction is driven, whereas a direction perpendicular to the drive direction should not be confined, such as direct drive XY-stages.

Apart from the air bearings, no precision engineered parts are needed. The accuracy is achieved using an advanced measurement system and feedback control.

Interesting for future research are: extension of the travel by expanding the stator, the use of magnetic instead of air bearings, and increasing the position accuracy and stability by modifications of the position measurement system.

References

- [1] E. Kallenbach *et. al.*; *Integrated Electrodynamic Multi-coordinate Drive ...*; Joint Hungarian-British Int. Mechatronics Conference; Budapest Sept. 1994, Proc. pp. 419ff.
- [2] F. Auer, *et. al.*; *Practical Application of a Magnetic Bearing ...*; Proc. 4th Int. Symp. on Magnetic Bearings, Hochschulverlag AG a.d. ETH Zürich, 1994. pp. 183-188
- [3] G.M. Bonnema; *Design of an XY-($\Phi=0$) Stage with Sub-micrometer Resolution*; Thesis of Post-Graduate Course in Technical System Design Engineering; Laboratory for Micro Engineering TU Delft; 1995.

A New Hybrid Concept for a Long Stroke Fast-Tool-Servo System

M. Weck, H. Oezmeral, K. Mehlkopp, T. Terwei
Fraunhofer-Institut für Produktionstechnologie, IPT
Aachen, Germany

Over the last years an increasing demand in fine structured ultra-precision parts with high surface and form quality can be noticed. Within the wide range of applications nonrotationally symmetric (nrs) high power laser optics are only one example for this. The use of lasers is more and more growing in industrial processing of different materials. An important aspect in this context is the necessity to adjust a specific intensity distribution for each application. This is usually realized by special optics, which are able to form the beam. To reduce the number of mirrors in beam guides of laser systems, special designed optics with nrs structures up to several 100 μm height are needed.

Based on the existing fast-tool-servo system at the IPT a new hybrid fast-tool-servo system, shown in [Figure 1](#), with two mechanically coupled actuators was designed: a piezo system with a stroke of 40 μm at 1000 Hz and a linear motor with a stroke of 2 mm at 40 Hz. The piezoelectric translator has a maximum force of 2400 N, a stiffness of 50 N/ μm and a resonance frequency with the loaded mass close to 2000 Hz. Three linear motors are connected in series and located symmetrically with the moving permanent magnets between the stators. Each stator is laminated with a single phase winding and the magnetic flux is created by the moving magnets. This structure of the motor allows to generate linear thrust forces with a high thrust/weight ratio and reduction of the normal force. The peak thrust force of the linear motor is 900 N with an electrical time constant of 5 ms.

The guides for the moving mass are designed by parallel springs. Their resonance frequency is located at 68 Hz, shown in [Figure 2](#). The total movement of the tool is measured by a laser interferometer with a resolution of 1nm. Additionally, the stroke of the piezoelectric translator is measured by a capacitive sensor and the speed of the linear motor is measured by a linear velocity transducer. The power loss of the linear motor and the piezoelectric translator is dissipated by a water cooling system. Temperature stabilisation is guaranteed by Peltier elements which control the heat dissipation to the cooling media.

The main problem of long stroke fast-tool-systems is the occurrence of the dynamic inertial forces during operation. It would not be possible to set up the fast tool system on a lathe without compensating the inertial forces. Several components of the lathe would be excited. A compensation of these forces to reduce exciting is given by the balancing of masses. Therefore, the stator of the linear motor is not fixed. Parallel springs allow a countermovement of the stator mass (s_{stator}) to the movement of the actor mass (s_{actor}). Due to the condition of balancing force and acceleration, the ratio of the masses is inverse to the amplitude rates ($m_{actor}/m_{stator} = s_{stator}/s_{actor} = 0,1$). The effective amplitude, which has to be actuated by the linear motor increases by 10%.

The servo-system of the Hybrid-Fast-Tool is not optimized yet. First measurements with position control system show that a further optimization is necessary because of the progressive spring characteristic. The closed loop frequency response for an amplitude of 500 μm (1 mm stroke of the linear motor) is shown in [Figure 3](#). It is possible to run the system up to 84 Hz without significant raise of the compliance. At 50 Hz a raise of nearly 0.3 dB can be detected. The response of the system on a square pulse with a stroke of 1mm is shown in [Figure 4a](#). After 10 ms rise time of the impulse an overshooting of 5% occurs. During the following transient state the noise was measured to 10 nm, [Figure 4b](#).

With the new long stroke Hybrid-Fast-Tool-Servo System a basis for manufacturing of nonrotationally symmetric surfaces up to a stroke of 2 mm is introduced. After optimizing at the control system it will be possible to manufacture a wide variety of optics. Additional problems are given by measurement of the surfaces and the long machining time for nonferrous metals, because of the limited cutting depth.

Keywords: diamond turning, actuators, fast-tool-servo, piezoelectric translator, linear motor, nonrotationally symmetric surfaces

References:

- /1/ Weck, M., Pyra, M.; Diamond turning of non-symmetric laser optics, Proceedings of the ASPE Annual Conference, 1990
- /2/ Miller, M.H.; Dow, T.A.; A controller architecture for integrating a fast tool servo into a diamond turning machine, Precision Engineering, January 1994
- /3/ Weck, M., Pyra, M., Oezmeral, H.; Non-Rotational-Symmetric Optics and their Applications, Proceedings of the 3rd International Conference on Ultraprecision in Manufacturing Engineering, May 1994
- /4/ Falter, K.J., Youden D.H.; The Characterization and Testing of a Long Stroke Fast Tool Servo, Proceedings of the 8th International Precision Engineering Seminar, May 1995

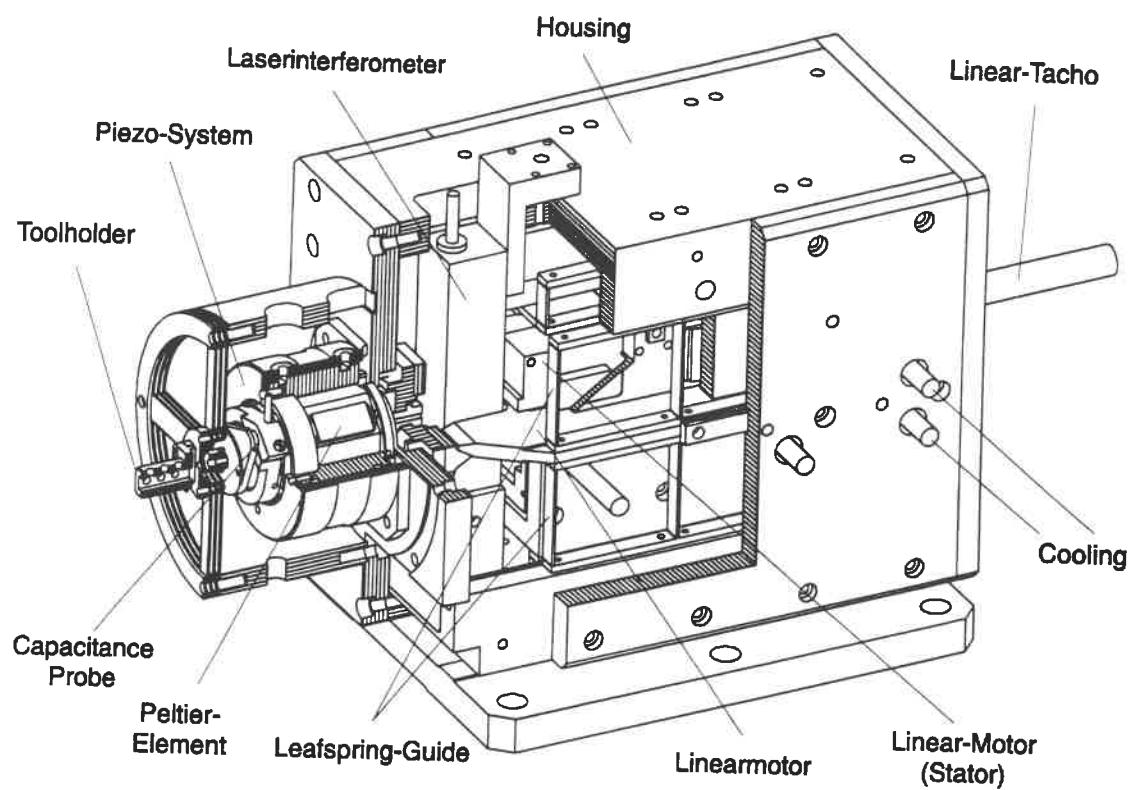


Figure 1: Design of the Hybrid-Fast-Tool

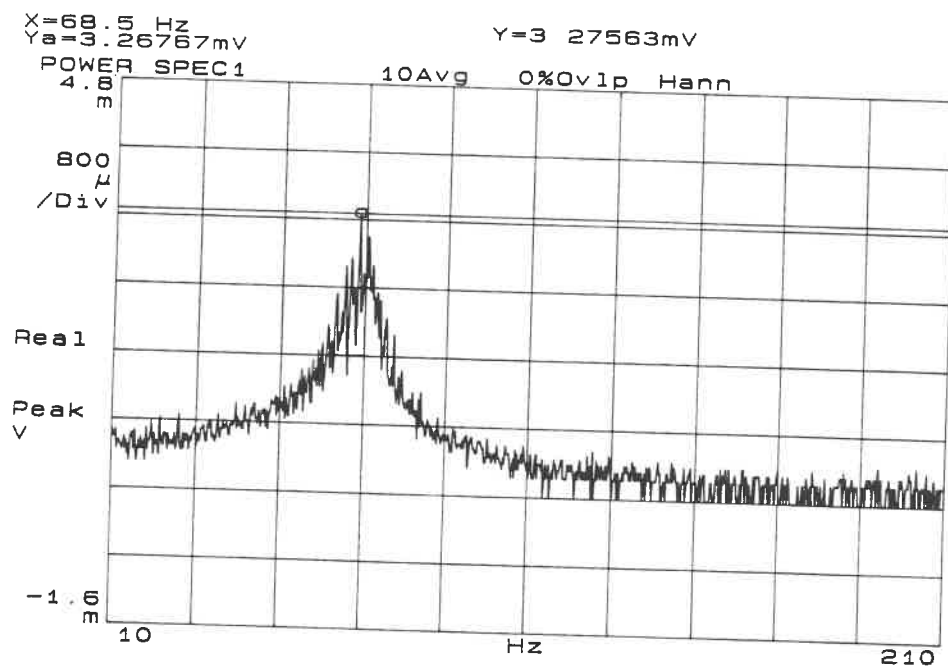


Figure 2: Mechanical resonance frequency

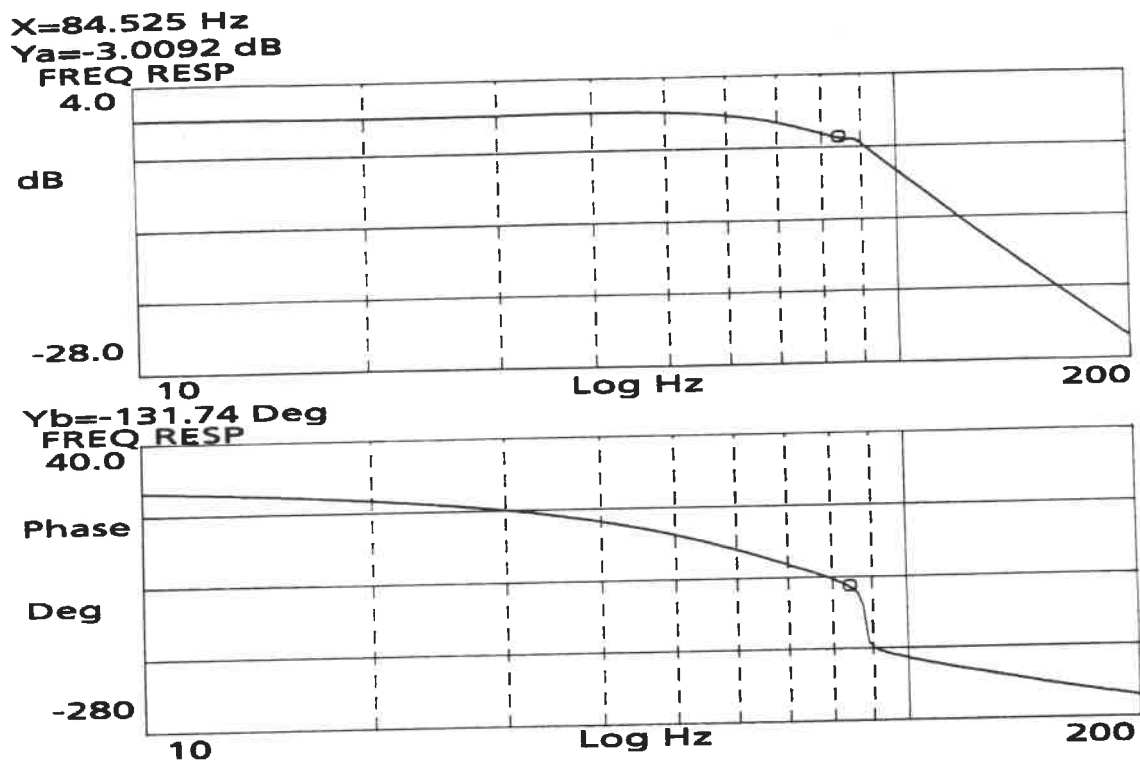


Figure 3: Frequency response (closed loop), amplitude = 500 μ m

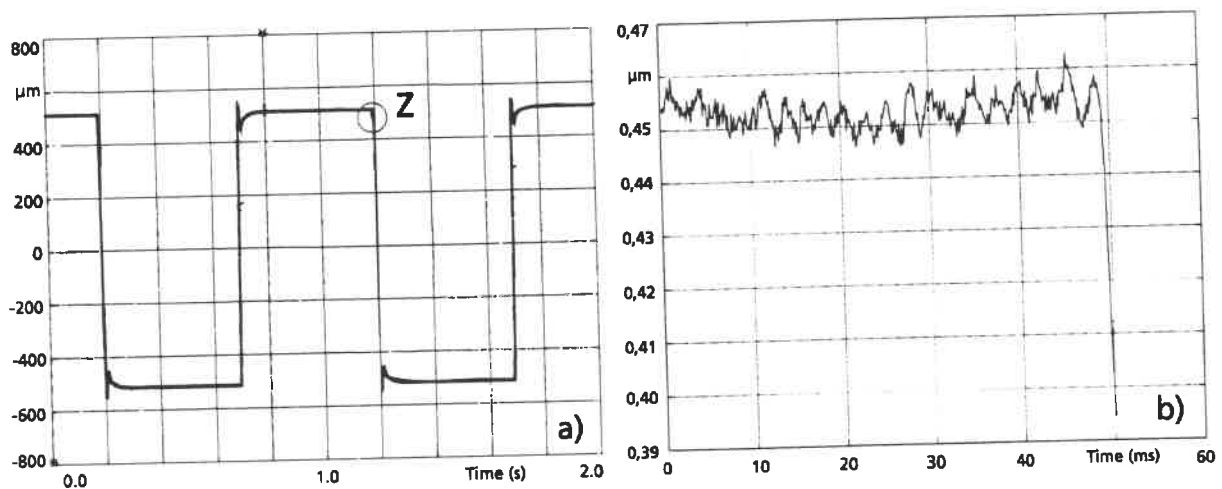


Figure 4: Transient response on a square pulse
 a) Amplitude = 500 μ m, frequency = 1 Hz
 b) Zoom in area Z

A New Inchworm® Motor For Industrial Nanopositioning Applications

David Henderson
Burleigh Instruments, Burleigh Park, Fishers, NY 14453

Summary

A new Inchworm motor concept (the Inchworm II) has been designed and demonstrated. This new design is expected to offer all the capabilities of the original Inchworm with many improvements which will make it a competitive high performance option for industrial applications. These improvements include better reliability, longer life, wider operating temperature range, higher speed, higher duty cycle and lower cost.

Original Cylindrical Clamp Inchworm

The Inchworm® is a piezoelectric (PZT) linear stepping motor that has been available commercially for over twenty years and achieves one nanometer resolution while providing hundreds of millimeters of travel. This unique combination of long travel and high resolution is achieved using three cylindrical PZT elements which move using a "clamp-extend-clamp" stepping motion. Push forces greater than 15 Newton's and speeds greater than two millimeters per second are routinely achieved. Other inherent characteristics of Inchworm motors include: high stiffness, zero backlash, low outgassing, non-magnetic materials and zero power dissipation when holding position.

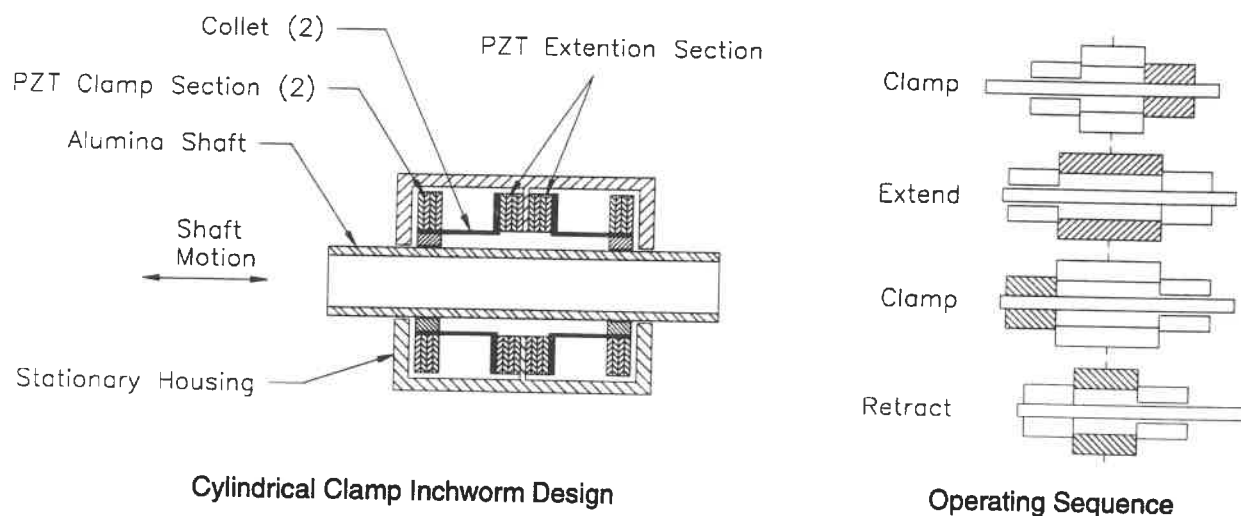


Figure 1- Original Inchworm Motor

Improvements Needed For An Industrial Inchworm

Burleigh's long experience with Inchworm technology has identified several areas where the design can be improved for industrial applications. These improvements include the following:

1. Eliminate or reduce tensile stresses in the PZT ceramic to extend life and reliability. A significant failure mode in the original Inchworm is cracking and arcing of the clamp elements

due to large cyclic tensile stresses (approaching 3000 psi at 1000 Hz). Tensile hoop stresses are inherent in the action of the cylindrical clamps as they grip the shaft.

2. Eliminate precision fits between clamps and shaft to reduce cost, extend the life, and support operation at temperature extremes. Cylindrical clamp Inchworms require an extremely close fit between the stationary clamps and moving shaft to operate properly. (Approximately a 0.25 micrometer tolerance.) This close tolerance fit is essential for converting the relatively small motion of the PZT (a few micrometers) into a large force. Achieving this precision fit requires significant time from highly skilled technicians to manually hone and match the Inchworm clamp to a mating shaft. The labor and yields of this process are a significant cost driver. In addition, the close fit is affected by wear and temperature changes which limits the Inchworm's life, speed, duty cycle, and operating temperature range.
3. Reduce the operating voltage to reduce cost and improve reliability and safety. The original Inchworm operates at 700 volts which places great demands on the electronics, cabling and motor packaging which are significant cost drivers. This high voltage is needed to produce significant strain in thick layered (0.75 mm) PZT material.
4. Achieve smoother velocity for constant feed and speed applications. Cylindrical clamp Inchworms produce a "glitch" or motion discontinuity each time the clamps are turned on and off. This glitch is caused by the cross coupling of the clamping motion. That is, the clamp is not a pure clamp but also changes length axially.

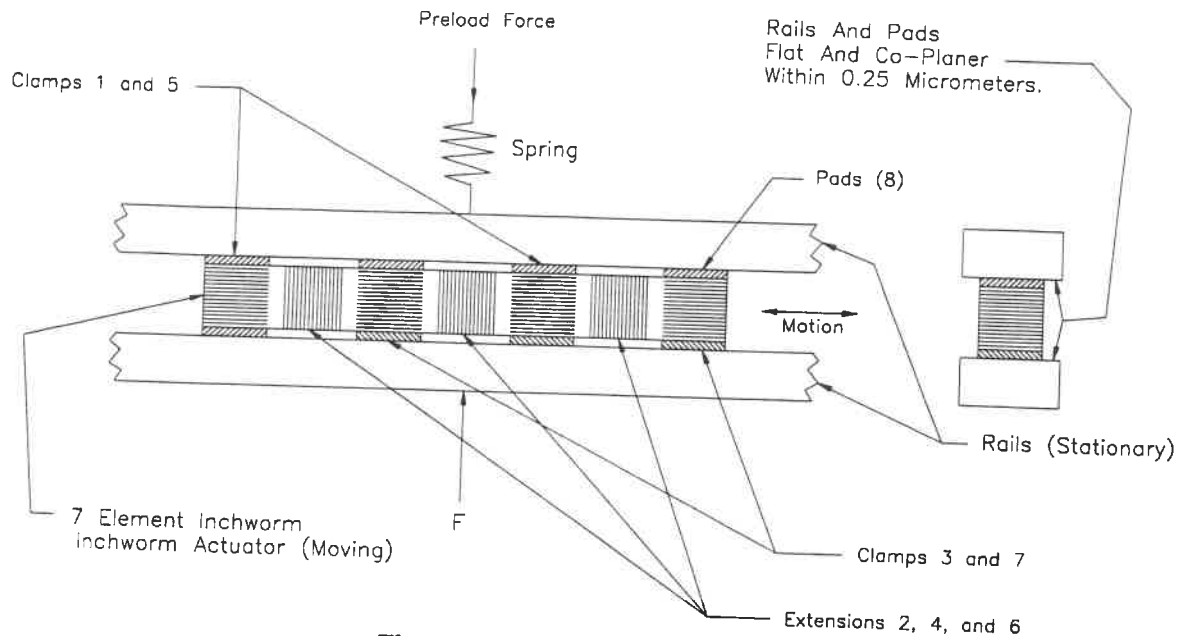
The Inchworm II Design

The Inchworm II design (Patent Pending) is shown in Figures 2, 3 and 4 and uses a seven section rectangular actuator. Four sections act as clamps and three sections as extension actuators. A four channel amplifier drives this actuator in a unique manner that eliminates the need for a precise clearance between the moving and stationary components. The moving and stationary elements can be held together with a constant preload force - much like ultrasonic motors but without the need to operate under resonant conditions.

The Inchworm II concept achieves all the improvements identified above for industrial applications.

1. Tensile stresses are eliminated using "inside out" PZT clamps that produce compressive stresses when activated.
2. Precision fits are replaced by spring loaded rails and a unique clamping motion that requires only flat and co-planar surfaces which are easily achieved by lapping. The actuators and rails are interchangeable and the spring loading compensates for wear and temperature changes. A spring loaded design also allows higher push force motors to be produced by simply increasing the preload and the size the PZT actuator.
3. Co-fired multilayer PZT actuators reduce the operating voltage by producing high strain at less than 100 volts. While the co-fired material (0.1 mm thick layers) does require higher current, the total system cost is reduced by reducing the voltage requirement.

4. A symmetrical design reduces the motion glitch associated with each clamp change. The attachment points on the seven element actuator are the centers of 2, 4, and 6. The relative spacing between these points does not change during motor operation. When a set of clamps is activated (1,5 or 3,7) the axial cross coupling motions of each PZT element are in opposite directions and essentially sum to zero.



**Figure 2 - Inchworm II Design
(Patent Pending)**

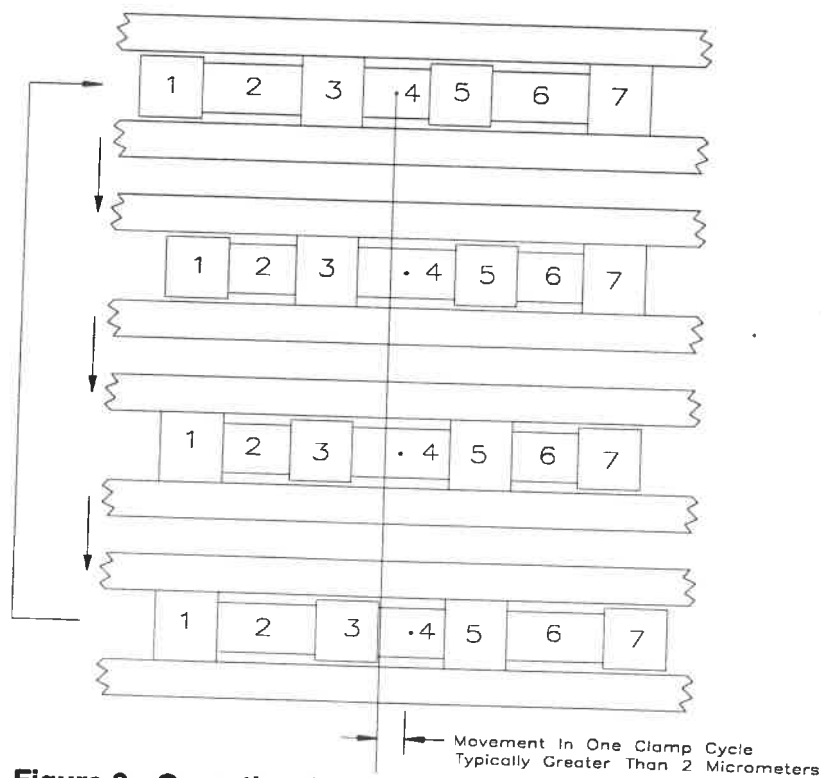


Figure 3 - Operating Sequence Of The Inchworm II Motor

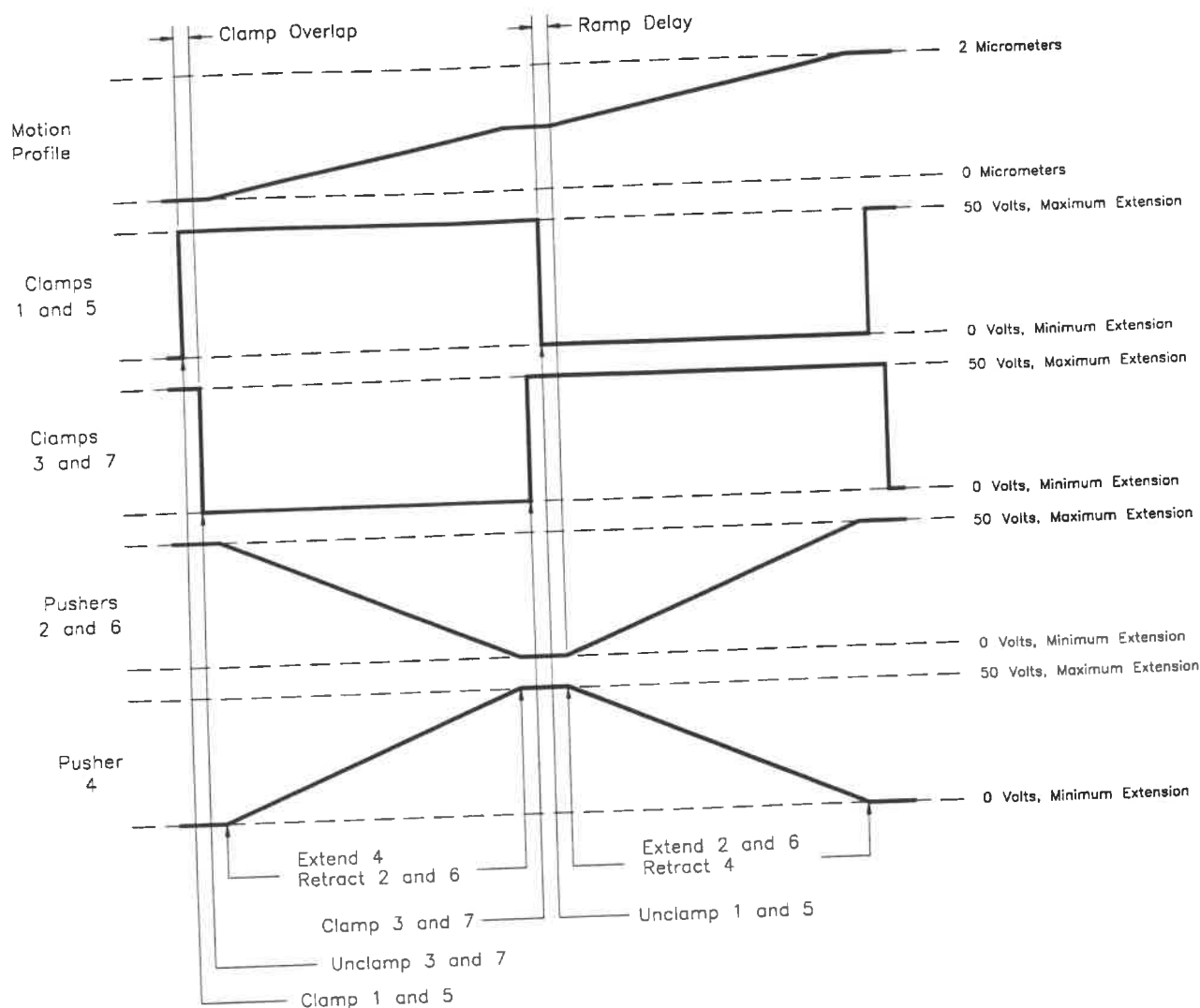


Figure 4 - Inchworm II Drive Signals and Motion Profile

Development Status and Challenges

The Inchworm II concept has been demonstrated and is currently being tested under a variety of operating waveforms (voltage, frequency, rise time, etc.) and preload conditions. Preliminary results are encouraging and much more data should be available at the conference.

Several technical challenges remain. A compact and simple method to apply preload for longer travel must be developed. Analysis of the load paths shows that the preload force must remain centered on extension segment 4 or the rails will rotate and no longer transfer the load between clamp pairs 1,5 and 3,7. Various carriages and bearing systems are being considered.

A cost effective method for manufacturing the seven element actuator must be developed. Epoxy bonding and co-firing of the seven element array are being evaluated. Particular care must be taken when connecting the seven elements to maximize axial stiffness while not significantly "clamping" the motion of adjacent segments.

Diagnostic of Diamond Turning Machining through FFT Analysis of Finished Surfaces

Arthur J.V. Porto¹, Jaime G. Duduch¹, Luiz C.R. Carpinetti¹, Anthony E. Gee², Renato G. Jasinevicius¹, Roberto Tsunaki¹

¹ Escola de Engenharia de São Carlos - USP
P.O. Box 359 - 13560-254 - São Carlos - SP- Brazil
² Cranfield University - Cranfield, Bedford, MK43 0AL, UK

Introduction

Although the surface roughness profile of machined surfaces can be theoretically reconstructed by the repetition the tool tip radius at intervals of tool-feed per revolution, experimental results show it as being highly dependent upon several factors, such as: cutting tool geometry, work-piece material properties, machining conditons, dinamic behavior and precision of the machine tool (Sata et al, 1986). It is possible to gain knowledge about the contribution of each of these factors on the finish of the machined suface through the analysis of the spacial frequency spectrum of the surface profile sampled by perfilometry techniques. Fast Fourier Transforms (FFT) is the mathematical tool for such an analysis (Pandit and Wu, 1983).

The ability to predict the influence of machining conditions, material properties and dynamic behaviour of the machine-tool on the texture of machined surfaces can help in reducing setting up times as well as improving the surface finishing quality and the economics of the machining process. With this in mind, an investigation was carried out so as to propose some guiding rules for diamond turning machining of different aluminium alloys and copper for optical applications.

Dynamic Behaviour of the Machine-Tool

The machine-tool under consideration was an ASG 2500 Rank Pneumo Aspheric Surface Generator. This machine has two axes of movement and is shown schematically in figure 1.

The part is held to the vacuum chuck mounted on the z axis. Several tests were made so as to identify the natural frequencies of vibration in the z and x directions. The frequency spectra were obtained by mounting an accelerometer on the vacuum chuck and on the tool fixture, and exciting the structure with a rubber mallet. Tests showed that one natural frequency of the spindle was 72 Hz and another of rigid body of around 1 Hz. The tool post was excited in the x and z direction. In the x direction, a spectrum of frequencies of 3.125, 7.0 and 84.5 Hz was observed, and in the z direction, 4.5, 7.5 and 15 Hz.

To measure the forced vibrations several tests were made as follows:

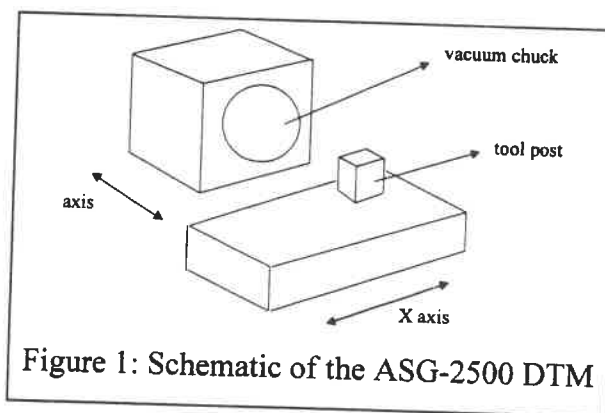


Figure 1: Schematic of the ASG-2500 DTM

- machine running with no movement of spindle and x and z carriages;
- spindle rotating at different speeds and x and z carriages stopped;
- x and z carriages moving at different speeds and spindle stopped;
- spindle and carriages moving at different speeds;
- machining test-pieces at various cutting conditions.

Figure 2 presents an example of the results. Generally, the results of the tests showed that the machine vibrates in its natural frequencies even when the spindle and carriages are at zero speed. As spindle speed varies from 600 to 1800 rpm, the amplitude of vibration diminishes. With the spindle stopped and the advances of the x and z axes varying from 5 to 50 mm/min, the amplitude of vibration in the x direction decreases. The amplitude of vibration of the natural frequencies in the x direction decreases substantially for advances smaller than 5 mm/min. However, this amplitude remains constant for higher values of advance. With the machine in cutting operation, the amplitudes of vibration in the natural frequencies assume higher values, and the higher the advance, the higher the amplitude of vibration in the natural frequencies. The effect of the cutting depth is not significant in the x and z direction. Finally, the increase of the spindle rotation for a constant feed rate of 5 $\mu\text{m}/\text{rev}$, has a non uniform effect, resulting in lower amplitudes for 1200 rpm and higher amplitude for a 1800 rpm.

The vibrations in the x direction, which is the direction of advance in face turning, change the spacing of the tool marks on the surface of the workpiece, since the tool advance will depend on the instant phase between feed rate spatial frequency and x axis vibration frequency. The ratio of wavelength (of the tool marks) to frequency of vibration is given by

$$\lambda = \frac{n a}{f_x} \quad (1)$$

where n is spindle rotational speed (rps), a is feed rate ($\mu\text{m}/\text{rev}$) and f_x is the frequency of vibration in the x direction.

The vibrations in the z direction, change the width and depth of the tool mark. Since the profile of the tool is reproduced on the surface of the workpiece in the form of a spiral, the displacement of the tool relative to the surface (in the z direction) causes a modulation of the profile of the surface in the radial (feedrate) direction. The wavelength of the modulated profile is given by (Takasu et al., 1985)

$$L = 2\pi \phi \frac{a}{n} \quad (2)$$

where ϕ is the decimal fraction of the ratio of the frequency of vibration to speed rotational speed (f_z/n), n is the rotational speed (rps) and a is the feedrate ($\mu\text{m}/\text{rev}$). When the phase shift per revolution of the workpiece is zero, the undulations will be aligned (in phase) and cannot be

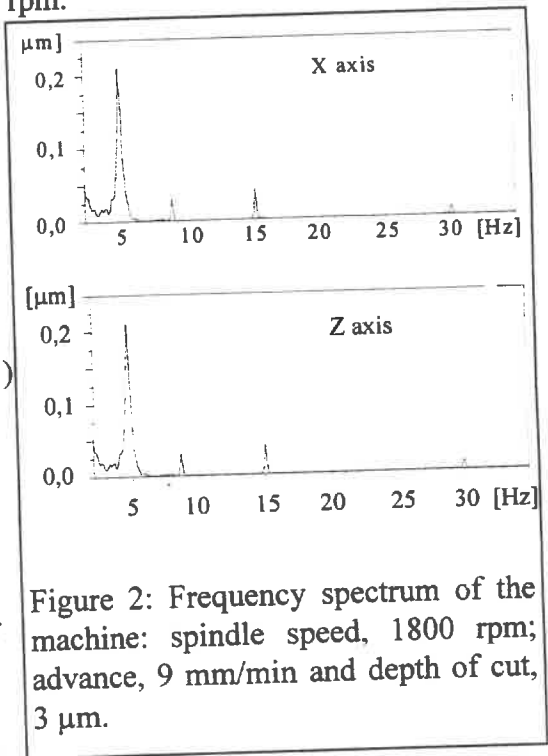


Figure 2: Frequency spectrum of the machine: spindle speed, 1800 rpm; advance, 9 mm/min and depth of cut, 3 μm .

detected by measuring in the radial direction. Thus, the components of the spatial frequency (generated by the natural frequencies and forced vibrations) of the roughness (measured radially) can be theoretically established. Application of these theoretical results indicates that the spatial frequencies range between hundredths and tenths of a cycle per micrometre. Simulation tests confirmed these results.

Machining Tests

Machining tests were carried out with the aim of identifying the nature and influence in the surface roughness of the following parameters: advance, cutting speed, depth of cut, workpiece material, geometry and sharpness of the tool. Single-crystal diamond tool with nose radius of 0.7 and 1.5 mm were used to machine copper 99.9% and three aluminium alloys (7075-T6, 6061-T6 and Kobe MB2). For each material and tool the feedrate was varied from 1 to 50 mm/min and the depth of cut from 1 to 15 μm . The surface roughness for the different materials was assessed keeping all machining parameters constant. The influence of the cutting speed was tested in tangential turning.

After machining the surface texture was measured over a 6 mm radial segment using a Form Talysurf¹ with a 0.08 mm cut-off roughness filter and an FFT algorithm was applied to the digitized profile data. Figure 4, for example, presents one of the frequency spectra obtained. Generally, the frequencies up to 0.1 cycles/ μm showed a constant average amplitude around 0.015 μm , and independent of the advance speed. Such results confirm that the lower spatial frequency roughness components would be caused by the natural frequencies of vibration and by the frequencies of forced vibrations. The spatial frequencies of the tool marks were well defined for feed rates above 10 $\mu\text{m}/\text{rev}$. For lower feed rates, the amplitude of low frequency undulations (machine vibration) were higher than the amplitude of the tool marks. The frequencies around 0.2 cycles/ μm , between 0.4 and 0.5 cycles/ μm and between 0.6 and 0.7 cycles/ μm had their amplitudes increased with the depth of cut in the region between 3 and 10 μm . However, for depth of cut of 1 μm the amplitude also had a significant increase, perhaps because of vibrations induced by chip rupture and/or difficulty in shearing hard inclusions. The lowest amplitudes were observed with the Kobe aluminum and copper (medium hardness). In tangential turning, higher cutting speeds caused higher amplitudes of vibration over all spectrum of frequencies.

In addition to the frequency spectrum analysis, the parameters R_a , R_q and Δq were assessed, showing significant increases for advances greater than 15 mm/min. With respect to the tool nose radius, no influence on the spatial amplitudes or roughness parameters could be observed. A microscopic analysis of the surfaces revealed some loss of reflectivity for extreme values of cutting depths.

Diagnostic Procedure

A diagnostic procedure of the machining process aims at optimizing the machining parameters so as to minimize the surface roughness. It comprises the following steps:

- a) setting machining conditions;
- b) plotting of the theoretical frequency spectrum based on the defined machining conditions;

¹ Rank Taylor Hobson Ltd, Leicester, UK

- c) analysis of the spectrum ;
- d) modification of the process based on the theoretical spectrum of the surface with the purpose of decreasing the surface roughness;
- e) machining of the workpiece;
- f) measuring the surface roughness
- g) plotting the frequency spectrum of the surface;
- h) analysis of the real spectrum
- i) optimization of the process.

The diagnostic procedure proposed here is divided into two phases, as follows:

- phase 1: definition of the components of the theoretical spectrum (steps *a* to *d*);
- phase 2: investigation of the type and degree of influence of the various aspects that affect the surface roughness (steps *e* to *i*).

Conclusions

The tests showed the feasibility of utilizing the proposed procedure for the optimization of the ultra-precision face turning operation. The analysis of the spectra from the dynamic and machining tests provided subsidies for the development of a theoretical model of the surface roughness which, besides the feedrate and tool nose radius effect, could also consider the dynamic behaviour of the machine, the depth of cut and the material. This procedure can be extended to other types of tools, materials and machine tools.

Finally, it can be concluded that the proposed procedure may be a useful tool to assess the machinability of different materials as well as to diagnose the influence of machining parameters, dynamic behaviour of the machine-tool and the cutting tool.

References

- Pandit, S. M. e Wu, S. M., Time series and system analysis with applications, John Willey & Sons, 1983.
- Sata, T. et al., "Analysis of surface roughness generation in turning operation and its applications", Annals of the CIRP, Vol. 34(1):473-476, 1985.
- Takasu, S.; Massuda, M. e Nishiguchi, T., "Influence of Steady Vibration with Small Amplitude Upon Surface Roughness in Diamond Machining", Annals of the CIRP, vol. 34 (1): 463-467, 1985.

Acknowledgements

The authors would like to thanks the FAPESP (Fundação de Amparo à Pesquisa do Estado de São Paulo) for the financial suport provided.

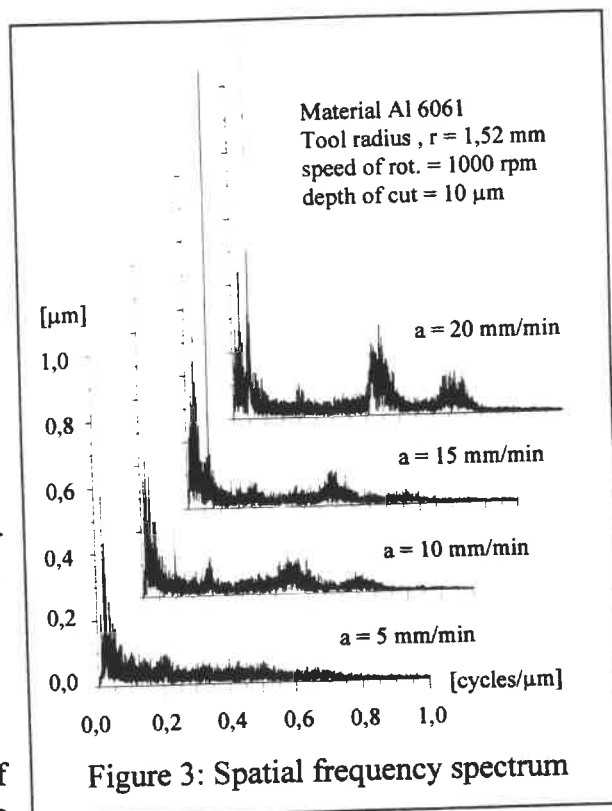


Figure 3: Spatial frequency spectrum

TEMPERATURE MODELING FOR ORTHOGONAL DIAMOND MACHINING

Balkrishna C. Rao, Robert X. Gao, and Craig R. Friedrich
Institute for Micromanufacturing
Louisiana Tech University
Ruston, LA 71270

INTRODUCTION

In recent years, the ultra precision cutting technique has been used extensively for the machining of small and delicate non ferrous workpieces to a high degree of surface finish and close tolerances. This is mainly due to the significant advances in machine tools and controls technology that make the technique less tedious. Despite its widespread utilization, the physics of micro machining at very small depths of cut is not well established due to the complexities of the deformation process. Although a lot of work has been done on the mechanics governing conventional machining or cutting at large depths of cut, there is a need to study the temperatures and forces produced in the micro domain. This paper presents the theoretical description of the temperatures encountered in micro machining. It considers the presence of a third source of heat at the flank-work interface that alters the heat transfer problem at very small depths of cut. To aid in the development of the mathematical expressions for the temperature field in the micro domain, this paper has developed the formulation of the tool-chip interface temperature for conventional orthogonal machining.

THERMAL ENERGY DISTRIBUTION IN MICRO MACHINING

In micro machining the distribution of the total energy dissipation is quite different from that of conventional machining. As depicted in Figure 1, the micro machining process involves significant sliding in the flank region in addition to shearing at shear plane and friction at rake face. In addition, because of the negative rake effect due to the roundedness of cutting edge, the tool-chip interface is divided into two regions. As illustrated in Figure 2, the nose of the tool forms the first region and the second region is the conventional tool-chip interface that begins at Q and ends at R. The presence of two distinct regions in the tool-chip interface is due to the shear plane that extends from the point that marks the intersection of nose and flank-work interface to the point of intersection of chip with workpiece. The additional heat generated due to flank work burnishing causes considerable conduction of heat between regions of tool-chip interface and flank-work interface. Besides transfer of heat between tool-chip interface and flank-work interface, there is certain heat transfer taking place between the individual regions of tool-chip interface.

CONVECTION BASED TEMPERATURE MODEL

In orthogonal cutting at large depths of cut, energy is converted into heat in the two principal regions of plastic deformation: shear zone and rake face friction zone [1]. Methods to determine the temperature in these zones of deformation based on principles of conduction heat transfer have

been developed by a number of workers [2]. The mode of heat transfer considered in this paper is that of convection. The "convection" terminology used may be misleading because in solids we are encountering problems of pure conduction [3]. However, from the point of view of an observer who rides on the diamond tool, the chip is a conducting medium that flows to the right with the chip velocity. Thus by solving the resulting energy equation subjected to certain boundary conditions through the Von Karmen-Pohlhausen integral method, an equation has been developed for the tool-chip interface temperature.

Referring to region PQ in Figure 3, and assuming that axial conduction is negligible and viscous dissipation is absent, the energy equation for forced convection boundary layer heat transfer for a constant property solid becomes

$$u \frac{\partial T}{\partial x} = \alpha \frac{\partial^2 T}{\partial y^2} \quad (1)$$

Here u is the horizontal component of fluid velocity, T is the temperature of any point in the chip domain, and α is the thermal diffusivity of the fluid. The temperature boundary conditions to which the above equation is subjected are:

$$T = T_t \text{ at } y = 0 \quad (2)$$

and

$$T = T_s + T_o \text{ at } y = \infty \quad (3)$$

where T_t is the temperature at tool-chip interface, T_s is the average temperature of shear plane, and T_o is the ambient temperature. The solution to the problem stated in equations (1-3) is obtained through the Von Karmen-Pohlhausen integral method [4-5]. Assuming a finite thermal boundary layer of thickness δ , Equation (1) can be solved with respect to y between 0 and δ , yielding

$$\frac{d}{dx} \int_0^\delta u T dy = \alpha \left. \frac{\partial T}{\partial y} \right|_0 \quad (4)$$

For slug flow limit the term u appearing in the above equation is replaced by the bulk chip speed V_c . After replacing u and further simplification, the above equation becomes

$$\frac{d}{dx} \int_0^\delta V_c T dy = -\alpha \left(\frac{\partial T}{\partial y} \right)_0 \quad (5)$$

Assuming similarity of temperature difference distribution through the thermal boundary layer, a function $\phi(\eta)$ is defined with

$$\eta = \frac{y}{\delta} \quad (6)$$

and

$$\phi(\eta) = \frac{\theta}{\theta_b} = \frac{T - T_t}{T_b - T_t} \quad (7)$$

Here ϕ is a function of single variable η , y is the distance of any point within the boundary layer from the tool face, θ and θ_b are temperature difference parameters, T is the temperature of the point at distance y from tool face, and T_b is the sum of T_s and T_o . Equation (5) can be written in terms of η and ϕ as

$$\frac{d}{dx} \int_0^1 \delta V_c (\phi(\eta) + \frac{T_t}{T_b - T_t}) d\eta = -\frac{\alpha}{\delta} (\frac{\partial \phi}{\partial \eta})_{\eta=0} \quad (8)$$

If T_t is expressed in terms of T_b and T is related to T_b , then Equation (8) becomes

$$\frac{d}{dx} \int_0^1 \frac{V_c \delta c}{1 - c} (1 - \phi(\eta)) d\eta = -\frac{\alpha}{\delta} (\frac{\partial \phi}{\partial \eta})_{\eta=0} \quad (9)$$

where c is a constant that expresses T as a definite percentage of T_b . According to the integral method a third degree polynomial is chosen to approximate the temperature distribution in the thermal boundary layer. i.e.,

$$\phi(\eta) = a + b\eta + c\eta^2 + d\eta^3 \quad (10)$$

Also,

$$\frac{\partial T}{\partial y} = 0 \text{ at } y = \delta \quad (11)$$

and

$$\frac{\partial^2 T}{\partial y^2} = 0 \text{ at } y = 0 \quad (12)$$

Using Equations (2-3) and (11-12), Equation (10) is solved to give

$$\phi(\eta) = \frac{3}{2} \eta - \frac{1}{2} \eta^3 \quad (13)$$

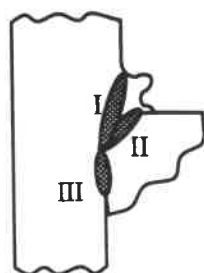
By modifying Equation (9) with the above equation, an expression is obtained for δ . On substituting this equation for δ in Equation (7), the formulation for T_t is obtained, which is:

$$T_t = \frac{T_b (\phi(\eta) - k_1)}{\phi(\eta) - 1} \quad (14)$$

where k_1 is the constant expressing T in terms of T_b .

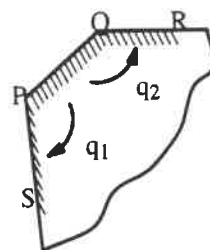
CONCLUSION

In this paper, the theoretical description of the distribution of the thermal energy in the micro domain has been presented. In order to develop a temperature model for the micro domain, this paper has first formulated an equation for the tool-chip interface temperature for conventional orthogonal machining. This equation, which is obtained from the principles of convection heat transfer, will be modified later for the micro domain. Currently, research is being conducted to modify the developed temperature model for the micro domain and its experimental verification.



- I Primary deformation zone
- II Secondary deformation zone
- III Flank-work interaction zone

Figure 1. Distribution of thermal energy in micro machining



- q_1 Heat flux conducted between PQ and PS
- q_2 Heat flux conducted between regions PQ and QR of chip-tool interface

Figure 2. Presence of two distinct regions in the chip-tool interface

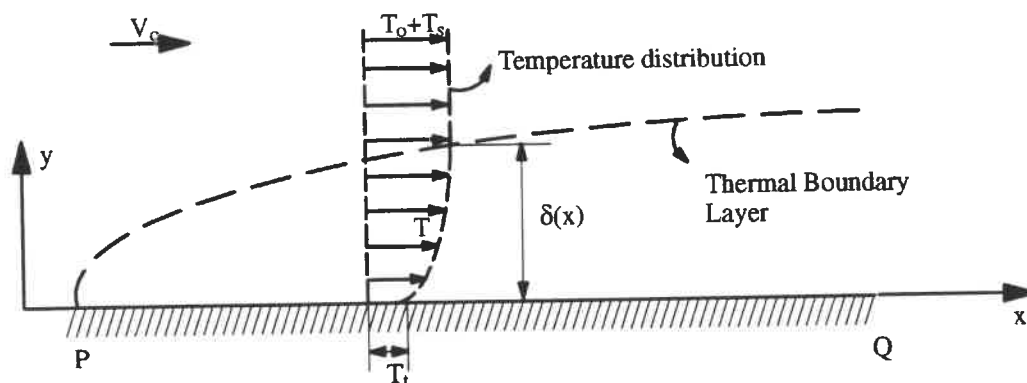


Figure 3. Convection model for region PQ of chip-tool interface

References

- [1]. G. Boothroyd, *Fundamentals of Metal Machining and Machine Tools*, Scripta Book Company, Washington, D.C., 1975.
- [2]. D.A Stephenson, "Assessment of Steady State Metal Cutting Temperature Models Based on Simultaneous Infrared and Thermocouple Data," *Journal of Engineering for Industry*, Vol. 113, pp. 121-128, 1991.
- [3]. P.A. Litsek and A. Bejan, "Sliding contact melting: The effect of heat transfer in the solid parts," *Transactions of the ASME: Journal of heat transfer*, Vol. 112, 1990, pp. 808.
- [4]. Schlichting, H., *Boundary layer theory*, New York: Mc Graw Hill Book Co., Inc., 1960.
- [5]. Biasi, L.C. "Transient convective heat transfer in fluids with vanishing Prandtl number: An application of the integral method," *International Journal of heat and mass transfer* Vol. 17, 1971, pp. 639.

Relationship between Catastrophic Fracture and Electric Resistance of Cutting Tools

- A trial for Catastrophic Tool Fracture Prediction -

Kunio Uehara Toyo University Kawagoe, Saitama, Japan
Naoki Obitsu Kiri Machine Mfg. Co. Ltd. Gunma, Japan

Summary: A clear correlation was observed between catastrophic fracture of cutting tool and the electric resistance of the tool point. This phenomenon was confirmed in machining carbon steel with P10 tool, cermet tool, conductive ceramic tool and coated tool whenever the catastrophic fracture tends to occur. An apparatus for the prediction of catastrophic fracture or the chipping of cutting tools was contrived and tested successfully.

Introduction

The catastrophic fracture or chipping of cutting tools is a still important problem in metal cutting field because it means a big loss of the manufacturing line when it occurs. However, it seems that no practical method for the prediction of the catastrophic fracture has been completed so far. The object of this study is to find out a suitable way for the prediction of the catastrophic fracture or chipping of cutting tools.

The fatigue or the deterioration of the tool material prior to its catastrophic fracture has been observed and studied by several researchers [1],[2]. Besides, according to a text book on the strength of materials [3], some physical properties such as electric resistance, hardness and dispersion character of ultrasonic waves are affected by the progress of the fatigue. These phenomena could be used for the prediction of the tool fracture. In 1989, the electric resistance between cutting tool and workpiece was tested for the prediction [4] and it was reported that the principle is correct but the S/N ratio is a big problem. In 1993, the acoustic emission in metal cutting was used for the prediction [5]. The former principle is adopted basically in this paper and the nature of the contact resistance between a tool tip and an electrode is examined in connection with the phenomenon of catastrophic fracture of cutting tools.

Experimental Setup

The fatigue or deterioration of tool material which causes the catastrophic fracture of cutting tool is generally localized to the tool point. Then, the electric contact resistance will be the most suitable value for the determination of the degree of deterioration. Fig.1 shows the

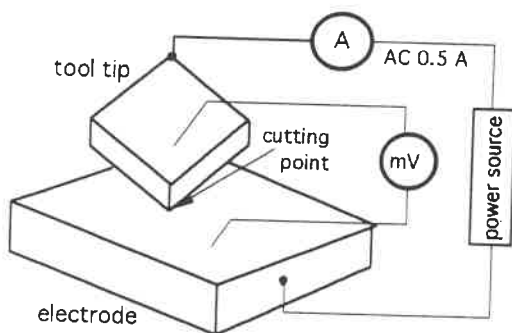


Fig.1 Principle of measurement

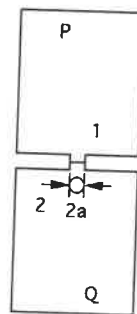


Fig.2 Model of point contact

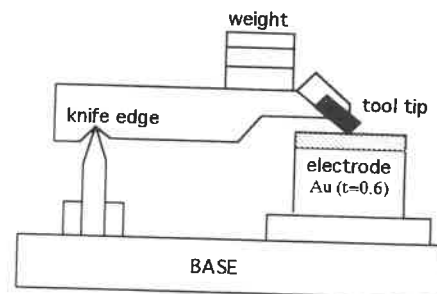


Fig.3 Measuring instrument

measuring principle of electric contact resistance used in this study and Fig.2 shows a model of point contact. The electric resistance between P and Q is mainly consisted of constriction and spreading resistance and it is expressed as follows:

$$R = (\rho_1 + \rho_2) / 4a \quad \text{--- (1)}$$

where ρ_1 and the ρ_2 are specific electric resistance of material 1 and material 2 respectively near the contact point. Then, measured electric resistance R should reflect the degree of the deterioration of tool material assuming the specific electric resistance of material is influenced by the progress of the deterioration. This is the basic concept of the study.

Fig.3 shows an experimental apparatus that makes sure the point contact between tool tip and flat electrode under constant load. A pure gold plate (0.6mm thick) was used as the electrode for to avoid the influence of oxide layer. The position of contact point is altered on the electrode in every measurement through sliding the electrode holder.

Experimental Results

Two kinds of carbon steel round bars (0.45%C and 0.55%C) which have four longitudinal grooves (5mm width) were turned with a high speed lathe. The cutting conditions are as follows: depth of cut: 1.0mm, 1.5mm, feed: 0.1mm/ rev. cutting speed: 95, 130, 135, 150, and 200 m/min. cutting fluids: none.

Five kinds of tool material (throwaway tip) were tested: P10, K10, Cermet(TiC,TaC type), Ceramic(Al_2O_3 ,TiC type) and Coated tool (TiN type), Type of tool tip: SNMN120408, Tool geometry: -6,-6,6,6,15,15,0.8mmr.

P10 Tool Fig.4 shows the change of the electric contact resistance with the progress of cutting. The mark X in the figure shows the occurrence of the fracture. The right hand pictures show the pattern of so fractured tool tips. The pictures A, B correspond to the position A, B in the graph. It is clear that the catastrophic fracture occurs when the contact resistance increases more than several times of the original value.

Cermet Tool Fig.5 shows the result of the experiment. The experiment was repeated four times and every data are shown in a figure. The mark X means the occurrence of the fracture and typical pattern of the fracture is shown in the right. It is recognized that the fracture occurs when the resistance reaches to a value that is from 30 to 40 times of the original value.

Conductive Ceramic Tool The result of four cutting tests is shown in Fig.6. The meaning of the mark X is the same as the Fig.5. The fracture tends to occur when the electric resistance increases more than three times of the original value. The fracture pattern is shown in the right.

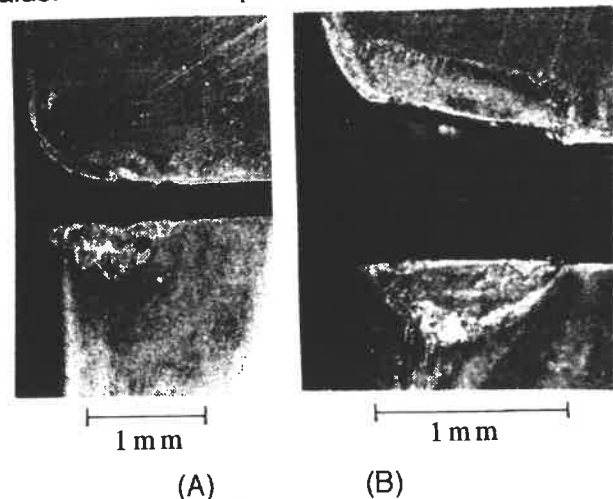
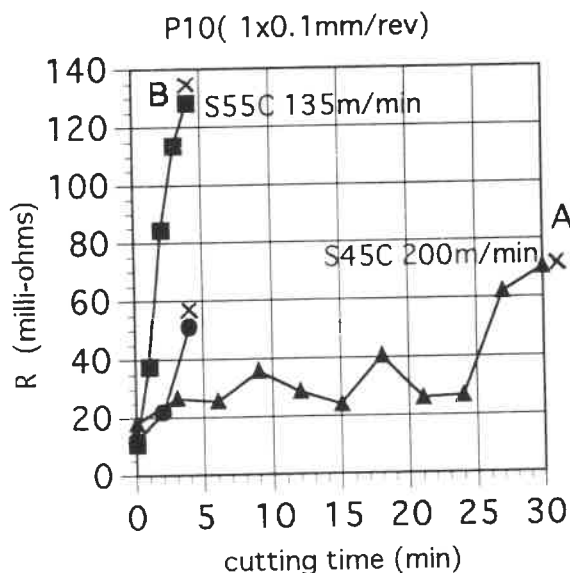


Fig.4 Contact resistance of P10 tool
workpiece : 0.45% & 0.55%C-steel
1mmx0.1mm/ rev. Contact load:2.6N

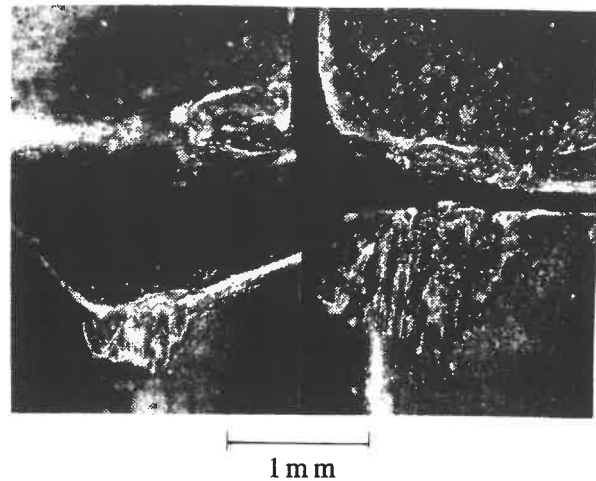
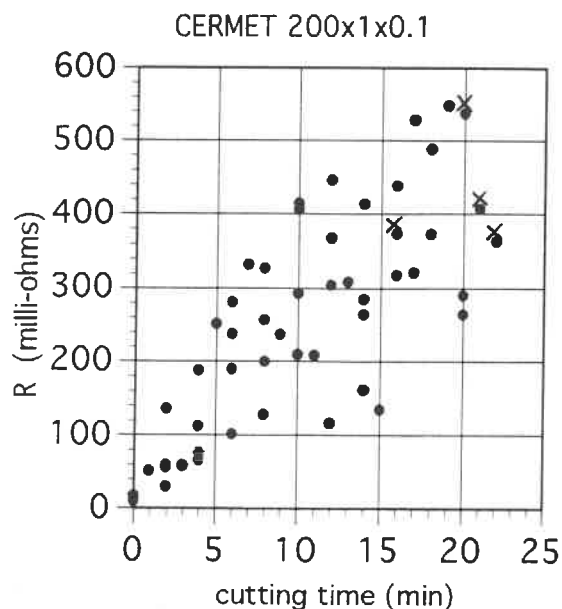


Fig.5 Contact resistance of cermet tool
workpiece : 0.45%C-steel, $V=200\text{m/min}$.
1mmx0.1mm/ rev. contact load:2.6N

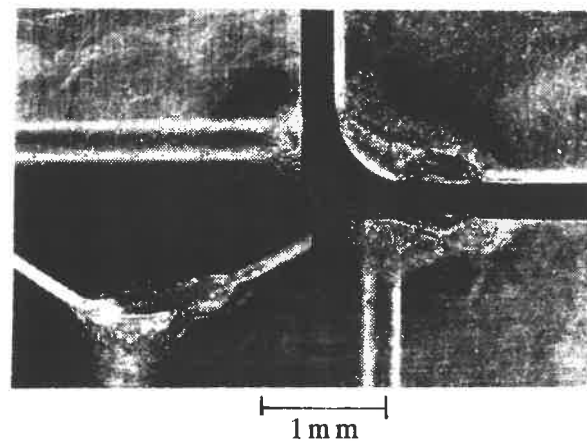
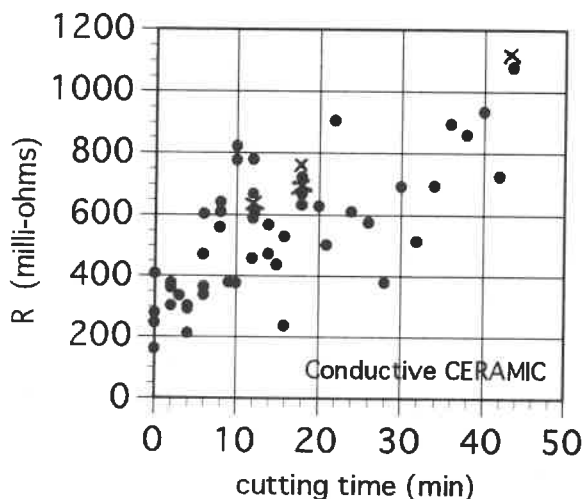


Fig.6 Contact resistance of ceramic tool
conductive ceramic (Al_2O_3 , TiC type)
workpiece : 0.45%C-steel, $V=200\text{m/min}$.
1mmx0.1mm/ rev. contact load:4.6N

Coated Tool The result of two cutting tests is shown in Fig.7. The same tendency as the former three tool materials is observed. The picture shows the fractured tool tip. However, a problem remains because the exfoliation of coated layer will affect the data. A lot of cutting test will be necessary in this case.

K10 Tool This is only tool material that showed no tendency of catastrophic fracture in this series of experimental study. A pure wear pattern was observed as is shown in the picture and the electric contact resistance did not increase in this case as is shown in Fig.8.

The results mean that the measuring method used in this study is sensitive to the catastrophic fracture of cutting tools and insensitive to the tool wear. This fact will be a strong proof that the electric contact resistance method is a suitable method for the prediction of catastrophic tool fracture. A sensing apparatus for the contact resistance measurement has been made and tested successfully based on the above findings. The detailed mechanism of the increase of the electric contact resistance of the tool point is a problem of the near future.

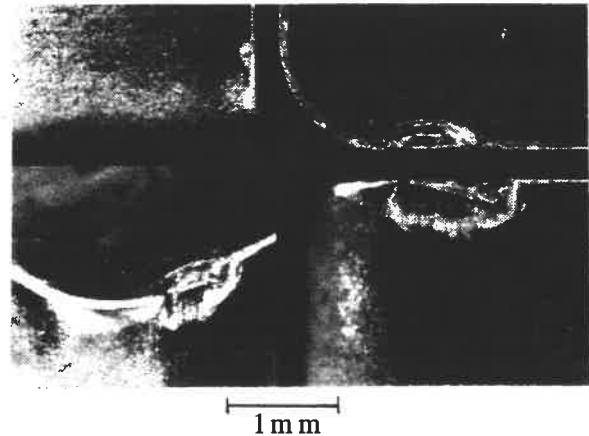
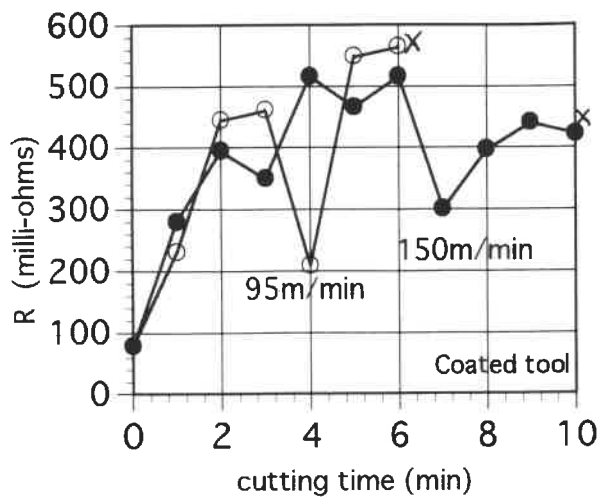


Fig.7 Contact resistance of coated tool
T530 coated tool (TiN type)
workpiece : 0.55%C-steel,
1.5mmx0.1mm/ rev. contact load:2.6N

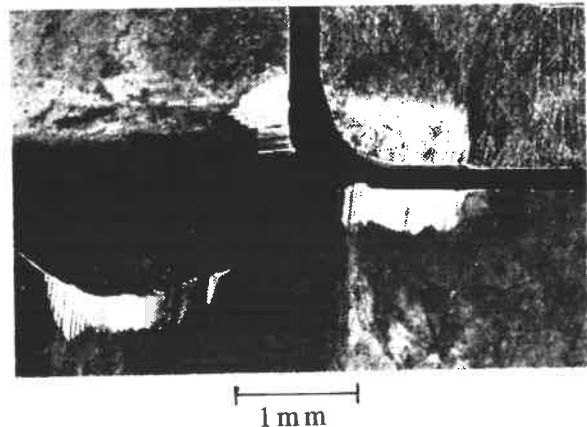
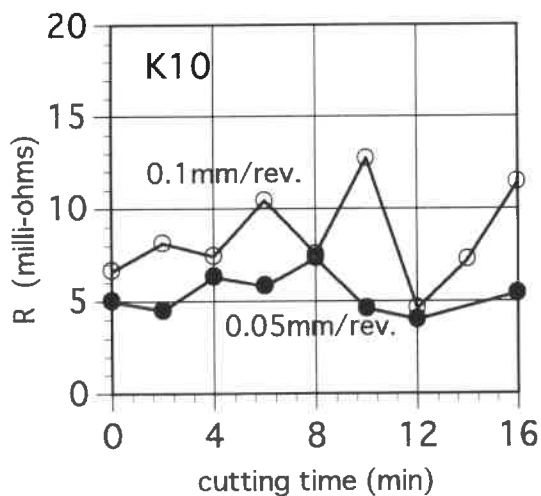


Fig.8 Contact resistance of K10 tool
workpiece : 0.45%C-steel, 130m/ min.
depth of cut:1mm. contact load:2.6N

Conclusions

The electric contact resistance between the nose point of cutting tool and the flat electrode was measured and examined in connection with the progress of cutting. The findings are as follows: (1) A clear correlation exists between the contact resistance and the deterioration of the tool material when the cutting tool tends to fracture. In this case, the electric resistance increases remarkably prior to the occurrence of the catastrophic fracture of the cutting tool. (2) The contact resistance does not change when the tool wear predominates and cutting tool shows no tendency of the catastrophic fracture.

The authors wish to acknowledge Mr. S. Arakawa and Mr. T. Yamazaki for their sincere cooperation of the experiment.

References

- [1] K. Uehara & Y. Kanda, Annals of the CIRP, Vol.25/1, 1977, p.11.
- [2] T. Shirakashi et al., Journal of the Japan Soc. for Precision Engg., 48-10, 1983, p.1348.
- [3] T. Yokobiri, Strength of Materials, 1955, Gihodo (Japanese Edition)
- [4] K. Uehara & H. Takeshita, Annals of the CIRP, Vol.38/1, 1989, p.95.
- [5] J.A. Rice & S.M. Wu, Trans ASME, Vol.115, November 1993, p.390.

INVERSE TECHNIQUE FOR ANALYZING THE ORTHOGONAL CUTTING PROCESS

by

Wei Xu*, Joseph Genin† and Qiwei Dong‡

Department of Mechanical Engineering
New Mexico State University
Las Cruces, NM 88003

Introduction

The first analytical study of cutting tool temperature in orthogonal cutting was given by Trigger and Chao [1]. Considering the tool tip as a stationary heat source and the heating at the chip contact surface as a moving heat source, they calculated the average interface temperature. Outwater and Shaw [2], Leone [3], Loewen and Shaw [4], and Chao [5] introduced some refinements to Trigger and Chao's calculations. But the question of temperature and heat-flux over the chip-tool interface was not answered.

Since the advent of the computer, finite element simulations of metal cutting are routinely used. For example, see Tay et al [6], Strenkowski and Carroll [7]. The problems mentioned previously still exist in their models. In [7], the contact length of the chip is measured by the wear scar on the tool. They assume that half this length is a sticking region and the other half a sliding region. In [8] the concept of a slide-line was introduced along the contact interface. Using this concept the authors of [8] were able to determine the contact length between the chip and tool as a function of time.

In this work we develop an inverse problem for the thermoelastic model to solve the contact problem in orthogonal cutting. Our goal is to determine the temperature distribution, thermal gradient, and the chip-tool contact length.

Problem Definition

Consider a two-dimensional domain Ω , shown in Fig. 1, bounded by the boundary Γ which may be written as $\Gamma = \Gamma_1 + \Gamma_2^*$. Here the boundary condition of Γ_1 is prescribed, the geometry of which depends on the geometry Γ_2^* . Γ_2^* is the unknown contact length with unknown boundary conditions.

The temperature distribution for the steady state condition is described by

$$\begin{aligned}\nabla^2 \phi(\bar{x}) &= 0; & \bar{x} \in \Omega \\ \phi(\bar{x}) &= \phi^0, & \bar{x} \in \Gamma_{1\phi} \\ \frac{\partial \phi}{\partial \bar{n}}(\bar{x}) &= q^0, & \bar{x} \in \Gamma_{1q}\end{aligned}\tag{1}$$

where $\bar{x}$ is the position vector, $\Gamma_{1\phi}$ and Γ_{1q} are the portions of Γ_1 with specified values of ϕ and q , respectively. The two inverse problems can then be written as follows.

I. Identification Problem: Here the contact length Γ_2^* is unknown. The set of experimental measurements available are

$$\begin{aligned}\phi(x_m, \Gamma_2^*) &= \phi_e^0(x_m, \Gamma_2^*), & x_m \in \Gamma_1 \\ q(x_n, \Gamma_2^*) &= q_e^0(x_n, \Gamma_2^*), & x_n \in \Gamma_1\end{aligned}\tag{2}$$

* Assistant Professor, † Professor, ‡ Graduate Student

where $m = 1, 2, \dots, M$; $n = 1, 2, \dots, N$. The subscript e represents the experimental measurements and the second variable Γ_2^* refers to these values as functions of the contact length. The identification problem consists of determining the contact length Γ_2^* based on these given experimental values.

II. Reconstruction Problem: Here the length of contact is known and the total boundary can be described as

$$\Gamma_\phi = \Gamma_{1\phi^0} + \Gamma_{2\phi}^*, \quad \Gamma_q = \Gamma_{1q^0} + \Gamma_{2q}^* \quad (3)$$

where the superscript 0 denotes specified boundary conditions, and the superscript $-$ denotes the unknown boundary conditions. The reconstruction problem consists of determining these unknown boundary conditions based on the prescribed boundary conditions given in Eq. (2). In the reconstruction problem, temperature measurements at interior locations of the tool may also be used. These measurements are expressed as

$$\phi(\bar{x}_p, \Gamma_2^*) = \phi_e^0(\bar{x}_p, \Gamma_2^*); \quad \bar{x}_p \in \Omega; \quad p = 1, 2, \dots, P \quad (4)$$

It is noted that the identification and reconstruction problems posed here must be solved simultaneously.

Theoretical Formulation

The boundary integral relationship, presented by Banerjee and Butterfield [9], for temperature in a solid corresponding to the actual contact area Γ_2^* is given as

$$\begin{aligned} \phi(\bar{x}, \Gamma_2^*) = & \int_{\Gamma - \Gamma_2^*} \left[\frac{\partial G(\bar{x}, \bar{y})}{\partial n(\bar{y})} \phi(\bar{y}, \Gamma_2^*) - G(\bar{x}, \bar{y}) \frac{\partial \phi(\bar{y}, \Gamma_2^*)}{\partial n(\bar{y})} \right] d[\Gamma(\bar{y}) - \Gamma_2^*(\bar{y}^*)] \\ & + \int_{\Gamma_2^*} \left[\frac{\partial G(\bar{x}, \bar{y}^*)}{\partial n(\bar{y}^*)} \phi(\bar{y}^*, \Gamma_2^*) - G(\bar{x}, \bar{y}^*) \frac{\partial \phi(\bar{y}^*, \Gamma_2^*)}{\partial n(\bar{y}^*)} \right] d\Gamma_2^*(\bar{y}^*) \end{aligned} \quad (5)$$

where G is a Green's function determined from

$$k \nabla^2 G(\bar{x}, \bar{y}) = \delta(\bar{x} - \bar{y}),$$

and, $\delta(\bar{x})$ is a Dirac delta function; $\bar{y}$ represents the coordinates of points which locate the boundary, and the superscript $*$ denotes quantities associated with the unknown contact area. To initiate the solution of the inverse problem, a contact area Γ_2 is assumed. Similar to Eq. (5), the temperature distribution for the assumed area Γ_2 can be written as

$$\begin{aligned} \phi(\bar{x}, \Gamma_2) = & \int_{\Gamma_1} \left[\frac{\partial G(\bar{x}, \bar{y})}{\partial n(\bar{y})} \phi(\bar{y}, \Gamma_2) - G(\bar{x}, \bar{y}) \frac{\partial \phi(\bar{y}, \Gamma_2)}{\partial n(\bar{y})} \right] d\Gamma_1(\bar{y}) \\ & + \int_{\Gamma_2} \left[\frac{\partial G(\bar{x}, \bar{y})}{\partial n(\bar{y})} \phi(\bar{y}, \Gamma_2) - G(\bar{x}, \bar{y}) \frac{\partial \phi(\bar{y}, \Gamma_2)}{\partial n(\bar{y})} \right] d\Gamma_2(\bar{y}) \end{aligned} \quad (6)$$

The variation of the temperature in the domain with an assumed contact length at the field location $\bar{x}$ with respect to the actual temperature is defined as

$$\delta\phi(\bar{x}, \Gamma_2) = \phi(\bar{x}, \Gamma_2^*) - \phi(\bar{x}, \Gamma_2) \quad (7)$$

The temperature $\phi(\bar{x}, \Gamma_2^*)$ is obtained from experimental measurements at location $\bar{x}$. Since the assumed contact area Γ_2 is known, the $\phi(\bar{x}, \Gamma_2)$ may be calculated using the usual boundary element procedure. Thus, $\delta\phi(\bar{x}, \Gamma_2)$ can be determined at the location $\bar{x}$. The right-hand side of Eq. (7) can be expressed by the integrals of Eqs. (5) and (6). The evaluation of the integral in

(5) requires using the contact length Γ_2^* , which is not known. We may express this integral in terms of the assumed contact length Γ_2 by using a Taylor expansion. For example, from Figure 1 we may write,

$$d\Gamma_2^* = d\Gamma_2 + \delta\Gamma_2, \quad \bar{y}^* = \bar{y} + \delta\bar{y} \quad (8)$$

Noting that

$$d\Gamma_2^* = |d\bar{y}^*|, \quad d\Gamma_2 = |d\bar{y}|$$

Then,

$$\phi(\bar{y}, \Gamma_2^*) = \phi(\bar{y}, \Gamma_2) + \frac{d\phi(\bar{y}, \Gamma_2)}{d\bar{y}} \cdot \delta\bar{y} + o(||\delta\Gamma_2||) \quad (9)$$

$$\phi(\bar{y}^*, \Gamma_2^*) = \phi(\bar{y}, \Gamma_2^*) + \frac{d\phi(\bar{y}, \Gamma_2^*)}{d\bar{y}} \cdot \delta\bar{y} + o(||\delta\bar{y}||) \quad (10)$$

$$= \phi(\bar{y}, \Gamma_2) + \frac{d\phi(\bar{y}, \Gamma_2)}{d\bar{y}} \cdot \delta\bar{y} + \delta\phi(\bar{y}, \Gamma_2) + o(||\delta\Gamma_2||, ||\delta\bar{y}||) \quad (11)$$

Boundary Conditions

The boundaries for Γ_2 and Γ_2^* may each be decomposed into two regions

$$\Gamma_2 = \Gamma_{2\phi} + \Gamma_{2q}, \quad \Gamma_2^* = \Gamma_{2\phi}^* + \Gamma_{2q}^*$$

The boundary conditions corresponding to Γ_2 and Γ_2^* are ϕ , ϕ^* , $\phi_{,n}$ and $\phi_{,n}^*$. First consider the unknown boundary conditions, $\phi(\bar{y}, \Gamma_2)$ and $\phi(\bar{y}^*, \Gamma_2^*)$, corresponding to $\Gamma_{2\phi}$ and $\Gamma_{2\phi}^*$. The value of $\phi(\bar{y}, \Gamma_2)$ on $\Gamma_{2\phi}$ is assumed. Define

$$\delta\phi(\bar{y}, \Gamma_2) = \phi(\bar{y}^*, \Gamma_2^*) - \bar{\phi}(\bar{y}, \Gamma_2), \quad \bar{y}^* \in \Gamma_{2\phi}^* \quad (12)$$

Then, using Eq. (11), Eq. (12) yields

$$\delta\phi(\bar{y}, \Gamma_2) = \delta\phi(\bar{y}, \Gamma_2) - \frac{d\phi(\bar{y}, \Gamma_2)}{d\bar{y}} \cdot d\bar{y}, \quad \bar{y} \in \Gamma_{2\phi} \quad (13)$$

Similarly, the unknown boundary conditions $\phi_{,n}(\bar{y}, \Gamma_2)$ and $\phi_{,n}(\bar{y}^*, \Gamma_2^*)^*$ can be expressed as

$$\delta\phi_{,n}(\bar{y}, \Gamma_2) = \phi_{,n}(\bar{y}^*, \Gamma_2^*) - \bar{\phi}_{,n}(\bar{y}, \Gamma_2) - \frac{d\phi_{,n}(\bar{y}, \Gamma_2)}{d\bar{y} \cdot d\bar{y}}, \quad \bar{y} \in \Gamma_{2q} \quad (14)$$

Here $\bar{\phi}_{,n}(\bar{y}, \Gamma_2)$ is the assumed value. Now substituting Eqs. (9), (11), (12) and (14) into the left-hand sides of Eqs. (5) and (6), and using Eq. (7) the formulation of the inverse heat conduction problem is obtained. Now the boundary element formulations are obtained by allowing the field point $\bar{x}$ to approach the boundary points on Γ_1 as well as any experimental measurement that may lie on the boundary Γ_1 or inside the domain. By numerical discretization, this may be written in matrix form as

$$[K]\{\hat{w}\} = \{f\}$$

The unknown quantities, $\hat{w}$, contain $\delta\phi(\bar{y}, \Gamma_2)$, $\delta\phi_{,n}(\bar{y}, \Gamma_2)$ and $\delta\bar{y}$. Using Eqs. (8), (12) and (14), we may determine the contact length Γ_2^* and the thermal boundary conditions $\phi(\bar{y}^*, \Gamma_2^*)$ and $\phi_{,n}(\bar{y}^*, \Gamma_2^*)$ using an iteration approach.

Numerical Results

The temperature distribution on a part of the tool is shown in Figure 2. The temperature and heat flux distributions on the rake surface are shown in Figures 3. and 4. The real contact length is 0.8mm, and the assumed contact length of the chip-tool interface was 1.2mm.

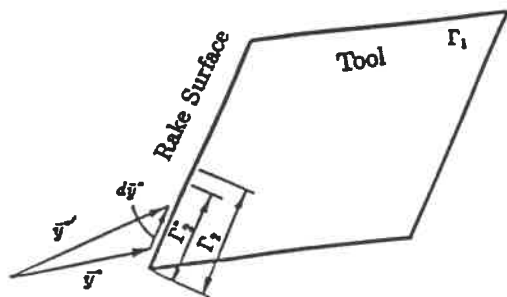


Figure 1: Two Dimensional Domain of Cutting Tool

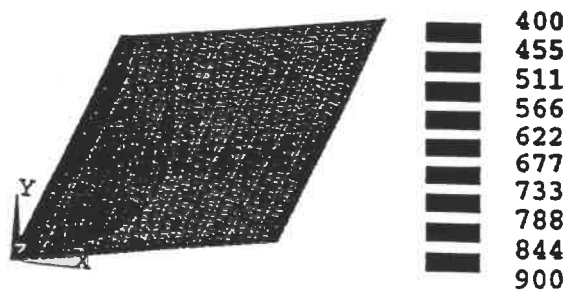


Figure 2: Temperature Distribution on Part of Tool

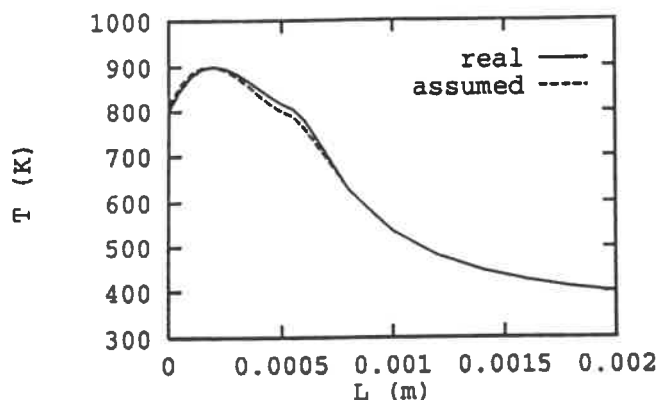


Figure 3: Temperature Distribution on Rake Surface

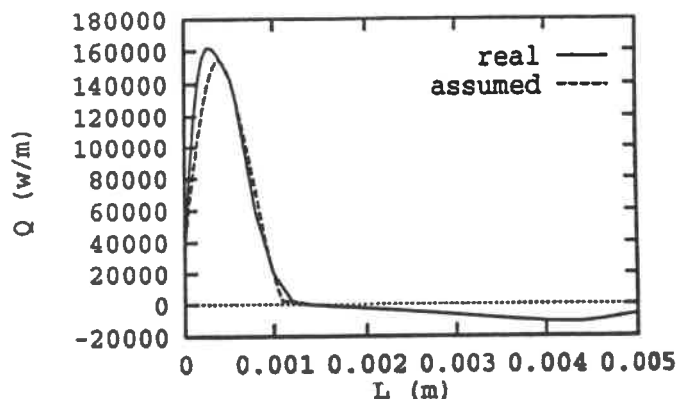


Figure 4: Heat Flux Distribution on Rake Surface

Acknowledgement

This research project was supported by the U.S. Army Research Office.

References

- [1] Trigger, K.J. and Chao, B.T., "An Analytical Evaluation of Metal Cutting Temperature", Trans ASME, vol. 73, P.57-68, 1951
- [2] Outwater, J.O. and Shaw, M.C., "Surface Temperatures in Cutting", Trans ASME, vol. 74, P.73-86, 1952
- [3] Leone, W.C., "Distribution of Shear-Zone Heat in Metal Cutting", Trans. ASME, vol. 76, P. 121-125, 1954
- [4] Loewen, E.G. and Shaw, M.C., "On the Analysis of Cutting-Tool Temperatures", Trans ASME, vol. 76, P. 217-231, 1954
- [5] Chao, B.T. and Trigger, K.J., "Temperature Distribution at the Tool-Chip and Tool-Work Interface in Metal Cutting", Trans ASME, vol. 80, P.311-320, 1958
- [6] Tay, A.O. et al, "Using the Finite Element Method to Determine Temperature Distributions in Orthogonal Machining", Proc. Instn. Mech. Engrs, vol. 188, P. 55-74, 1974
- [7] Strenkowski, J.S. and Carroll, J.T.III, "A Finite Element Model of Orthogonal Metal Cutting", J. Engineering for Industry, vol. 107, P. 349-354, 1985
- [8] Laake, B.A. and Anderson, C.A., "Finite Element Modeling of Orthogonal Machining Using an Elastic-Viscoplastic Material Model", Los Alamos-12151-MS, UC-906, 1991
- [9] Banerjee, P.K. and Butterfield, R., "Boundary Element Methods in Engineering Science", McGraw-Hill, U.K., 1981

CHIP SEGMENTATION IN MACHINING AISI 52100 STEEL

Matthew A. Davies, Chris J. Evans, Kari K. Harper
Manufacturing Engineering Laboratory
National Institute of Standards and Technology
Gaithersburg MD 20899

Introduction

The availability of polycrystalline cubic boron nitride (CBN) tools has made the single-point turning of hardened steels a viable alternative to grinding for the production of precision parts such as rollers for bearings and injection molds. However, factors such as rapid tool wear and undesirable deformation and vibration of the machine tool still severely limit the quality of surfaces produced by this process. In this work, we examine chip formation in the machining of 52100 bearing steel. A transition from continuous to segmented chips is found to occur as velocity and depth of cut are increased. Chip segmentation does not appear to be due to brittle fracture^{1,2} but instead is consistent with the model for catastrophic thermoplastic shear first proposed by Recht³. However, the transition from nonsegmented to segmented chips is not abrupt. Rather it seems to be characterized by a set of intermediate parameter values over which segments occur at closely spaced but somewhat irregular intervals while at higher parameter values segment spacing becomes larger and more uniform. Although similar trends have been depicted in the literature⁴, no model has been proposed to explain the apparent complexity of this transition. This work marks the beginning of an effort to understand and model material removal in hard turning and to correlate the characteristics of the material removal to important parameters such as surface roughness, residual stress and rate of tool wear.

Experimental Set-Up

Experiments were conducted on a precision air-bearing spindle with a 1.5 kW motor mounted to a two-axis precision machine base. The work material was 52100 bearing steel with a hardness of 62 R_C. Orthogonal cuts were performed on the edge of a ring of material 0.5 mm wide with a diameter of 94 mm using custom ground and lapped CBN tools. The set-up is shown schematically in Figure 1 where v is the cutting velocity, t is the uncut chip thickness and α is the rake angle. Chips were collected for a variety of different cutting parameters, mounted flat with rake face down, and photographed in a Scanning Electron Microscope (SEM). These photographs were used to measure the spacing, d , between segments in the chips. SEM measurements of chip segmentation were verified using a stereo optical microscope.

Results

In this work, the change in the average distance between the shear zones of segmented chips was evaluated as a function of two parameters, cutting speed and depth of cut.

The dependence of shear zone spacing on cutting speed was investigated using the orthogonal cutting conditions described above. For these experiments, the uncut chip thickness was held constant at 33 μm , the rake angle was -25° , and the cutting speed was varied from 0.35 m/s to 3.1 m/s. At each value of the cutting speed, the average spacing of twenty consecutive chip segments was measured for ten chip samples. The average and estimated standard deviation of these measurements were then calculated. Figure 2 shows SEM photographs of chips cut at four different speeds. At 3.1 m/s, the chips appear highly segmented. The mean size of the chip segments is 58 μm and they appear extremely

regular. At a cutting speed of 0.97 m/s the chips still appear segmented, but the mean segment size has decreased to $25\text{ }\mu\text{m}$ and the segments have become more disordered, often not extending across the entire width of the chip. At a cutting speed of 0.35 m/s the material flow appears nearly continuous with shear laminae spaced at an average of $2\text{ }\mu\text{m}$ intervals. Figure 3 shows a plot of the average distance between shear zones as a function of cutting speed. The data suggests a sharp increase in average shear segment spacing at a critical speed of about 0.4 m/s. At higher speeds, the curve appears to flatten out and approach some limiting segment spacing. The general shape of the curve indicated by the data is consistent with that proposed by Recht³ and suggests that chip segmentation is due to catastrophic thermoplastic shear. Other materials exhibit similar behavior, but the critical speeds for the onset of thermoplastic shear vary widely. For AISI 4130 steel, a critical speed of 4.5 m/s has been reported⁵. For nickel-iron base super alloys this speed is lower⁴, approximately 1 m/s, while titanium is segmented for nearly all cutting speeds⁶.

The dependence of chip segmentation on uncut chip thickness was also investigated using orthogonal cutting operations. Figure 4(a) shows a plot of shear zone spacing versus uncut chip thickness for two different rake angles, 0° and -25° degrees, and a cutting speed of 1.45 m/s. In both cases, the size of the chip segments varies strongly with uncut chip thickness. This is strikingly evident in Figure 4(b) which shows a chip obtained from a facing operation. This chip was cut by a $394\text{ }\mu\text{m}$ nose radius CBN tool with a rake angle of approximately -25° and a cutting speed of 2.65 m/s. The feed rate was $15\text{ }\mu\text{m}$ per revolution and the depth of cut was approximately $250\text{ }\mu\text{m}$. The thickness of the chip formed under these conditions varies from approximately $12\text{ }\mu\text{m}$ at the left edge, where the chip appears highly segmented, to nearly zero at the right edge, where it appears continuous. This suggests that measurements of segment spacing in orthogonal cutting may be used to understand and predict chip morphology in more practical cutting operations such as facing.

Discussion

Two major conclusions can be drawn from this work. First, the dependence of average shear zone spacing on velocity is consistent with the model proposed by Recht, and therefore the mechanism for chip segmentation appears to be thermoplastic shear instability in the material flow. This instability has been documented in numerous materials and is due to the dominance of thermal softening effects over strain hardening effects in the shear zone. However, there are no models to explain the irregularity in the spacing between shear zones observed at speeds just above the critical value for the onset of segmentation. One possible explanation for this behavior is that the heat generated in the formation of one zone can soften the material ahead of the tool and effect the location of the next zone. This would imply a dynamic interaction between the thermal and mechanical problems which is ignored in Recht's simplified model. The second major conclusion that can be drawn from this work is that chip segmentation frequency varies strongly with depth of cut, as evidenced in chips with varying thickness.

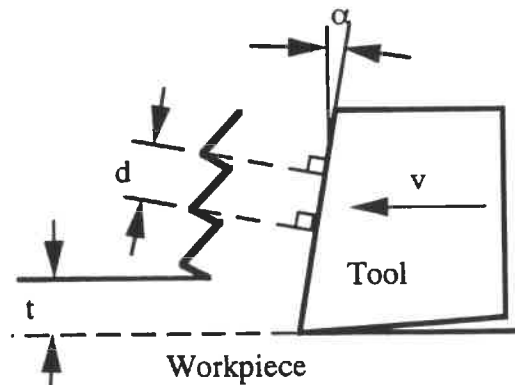
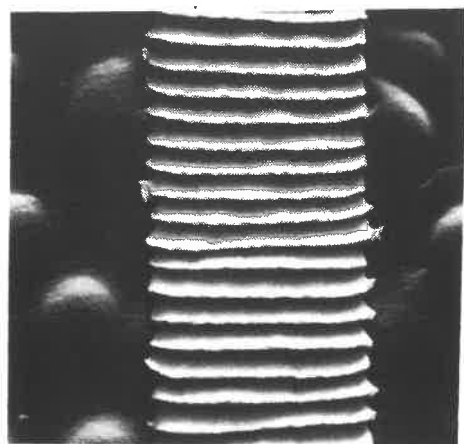


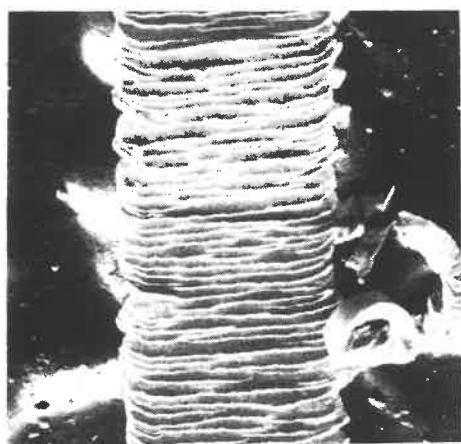
Figure 1 - Orthogonal Cutting Schematic



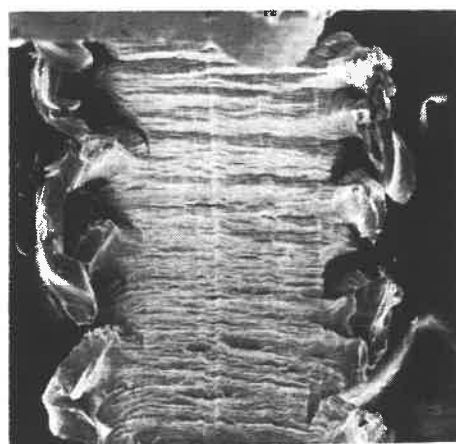
(a) 3.1 m/s



(b) 1.5 m/s



(c) 0.97 m/s



(d) 0.35 m/s

400 μm

Figure 2: Chips formed at various cutting speeds.

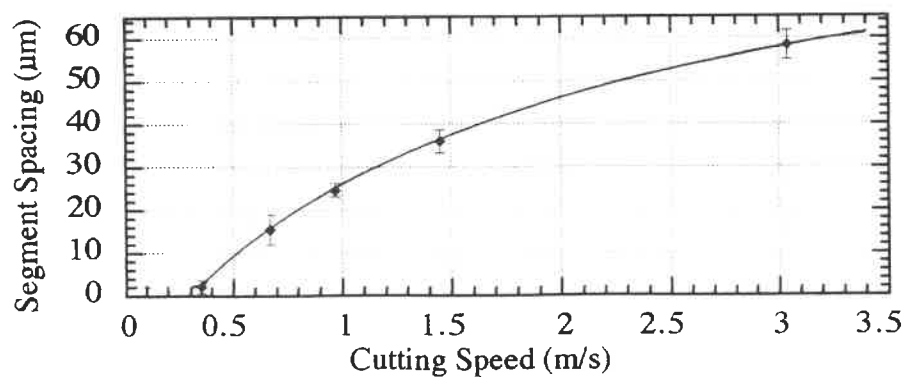
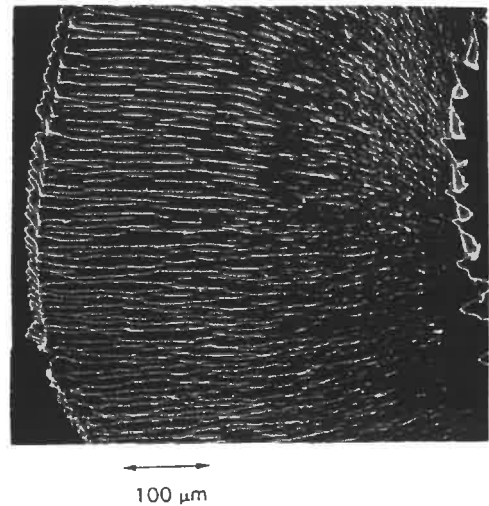
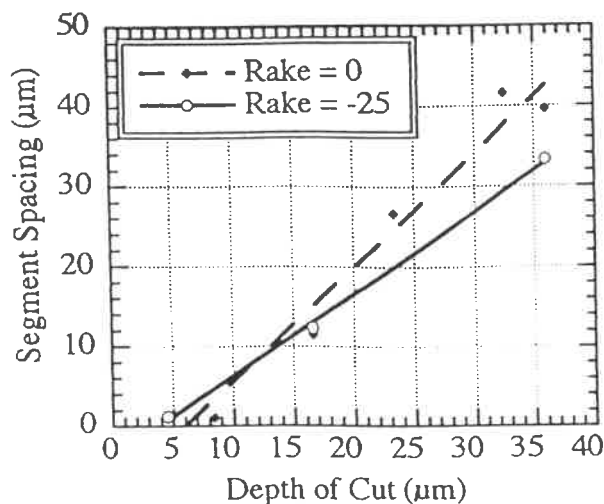


Figure 3 - Plot of mean spacing between shear zones as a function of cutting velocity. Error bars are based upon a coverage factor of 2.



(a)

(b)

Figure 4 - (a) Chip segment spacing versus depth of cut. (b) Chip formed in a facing operation.

Future work includes an examination of polished and etched chips to determine if thermoplastic shear is the mechanism for chip segmentation at all values of the cutting parameters. These efforts will also help to study the phenomena underlying the apparent disorder in the shear zone spacing during the transition to segmented chips.

References

- [1] M. C. Shaw and A. Vyas, "Chip Formation in the Machining of Hardened Steel," *Annals of the CIRP*, (1993) **42**(1), 29-32.
- [2] W. König, M. Klinger and R. Link, "Machining of Hardened Steels with Geometrically Defined Cutting Edges - Field of Applications and Limitations," *Annals of the CIRP*, (1990) **39**(1), 61-64.
- [3] R. F. Recht, "Catastrophic Thermoplastic Shear," *Journal of Applied Mechanics*, (June 1964), 189-193.
- [4] R. Komanduri and T. A. Schroeder, "One Shear Instability in a Nickel-Iron Base Superalloy," *Journal of Engineering for Industry*, (1986) **108**, 93-100.
- [5] R. Komanduri, T. Schroeder, J. Hazra, B. F. Von Turkovich and D. G. Flom, "On the Catastrophic Shear Instability in High-Speed Machining of an AISI 4340 Steel," *Journal of Engineering for Industry*, (1982) **104**(2), 121-131.
- [6] R. Komanduri, "New Observations on the Mechanism of Chip Formation when Machining Titanium Alloys," *Wear*, (1981) **69**, 179-188.

Grinding mechanisms of resinoid bonded diamond wheels with ultra fine grains

Jianshe Tang*, Katsuo Syoji**

David Dornfeld* and Tsunemoto Kuriyagawa**

*Department of Mechanical Engineering

University of California at Berkeley, Berkeley, CA94720, USA

** Department of Mechatronics and precision Engineering

Tohoku University, Sendai 980, JAPAN

Abstract

Compared to lapping and polishing, which are commonly used in super finish surface machining of brittle materials, ultra precision grinding employing diamond wheels with ultra fine grains is an attractive method because high form accuracy is achievable at high material removal rates. For a comprehensive understanding of the grinding mechanisms of this type of diamond wheel, this research experimentally investigated the influence of two fundamental grinding parameters, wheel-workpiece contact length l_c and maximum grain depth of cut g_m , on grinding forces. The variation in shape of grains on a diamond wheel surface in grinding process was also observed with a SEM for different grinding conditions.

1. Introduction

Manufacturing of resinoid bonded diamond wheels with ultra fine grains is one of the key technologies developed to meet the requirements of ultra precision grinding of ceramics. In the past 5 years, research on the applications of this type of diamond wheel for super finish grinding has been extensively carried out. Abe et al. [1] built an ultra precision straight surface grinding machine with a loop stiffness of 150 N/ μ m and feed resolution of 10 nm. By using this machine and employing resinoid bonded diamond wheels with ultra fine grains, ductile mode grinding of silicon wafer was done. Zhou and Syoji [2] developed a truing and dressing device, namely the GC cup truer, for dressing this type of wheel. Optimal dressing conditions were also given in their paper. In addition, Ichida et al. [3] investigated the optimal grinding conditions for resinoid bonded diamond wheels with ultra fine grains from the point of view of grinding ratio. This project studies the grinding mechanisms, that is, what happens as abrasive grains interact with the workpiece.

There are a number of approaches, such as direct observation of the wheel surface, force and acoustic emission (AE) signal sensing and analysis [4, 5], which are effective for gathering evidence to identify the mechanisms of abrasive-workpiece interaction. In this paper, we analyze wheel surface and force variation in grinding. Work on correlating the AE signal to grinding mechanisms is underway in the Laboratory for Manufacturing Automation in UC Berkeley.

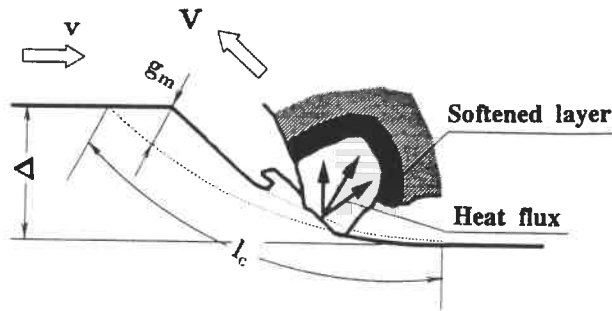
2. Experimental method and conditions

We can take the grinding process as a collective effect of individual grain cutting action as shown in Figure 1. Then g_m and l_c should be the two decisive parameters for the grinding process, because g_m represents a heat flux flowing into a grain or represents the force acting on a grain, while l_c represents the time of a grain passing through the grinding zone. In this study, as one aspect of grinding mechanisms, we investigate the influence of g_m and l_c on grinding forces and grain variation on a wheel surface in grinding.

As shown in the two expressions in Figure 1, if we change table speed v while keeping wheel depth of cut Δ constant, we can get different g_m values for the same l_c . So as an experimental method, at first we evaluate the influence of g_m on grinding performance by varying table speed v ; then we discuss the influence of l_c by selecting several wheel depths of cut Δ and repeating the same experiments.

Table 1 Grinding conditions

Wheel		SD1500P150B7 SD1500R150BRC SD1500-150MT
Truing	Truer	GC#800J~3000J
	Truer speed	200m/min
	Wheel speed	380m/min
	Table feed rate	0.4m/min
	Depth of cut	1.0 μ m/pass
Grinding	Wheel speed	1800m/min
	Work speed	0.4m/min~8m/min
	Infeed rate	0.5 μ m ~ 8 μ m



$$l_c = (D\Delta)^{1/2}, \quad g_m = 2\alpha v/V(\Delta/D)^{1/2}$$

α : Length between successive cutting grain

Fig. 1 Model of a grain in cutting

Grinding operations were carried out on an ultra precision surface grinding machine whose feed resolution is 0.1 μ m. With average grain diameter of 10 μ m, three pieces of diamond wheels, resinoid, resinoid/metal composite and metal bonded wheels are used. Each of them has a removable segment, so we can use the SEM to observe the morphology of a diamond grain on the wheel surface in a specific stage of the grinding process. Truing and dressing are performed with a GC cup truer[2]. For different bond materials, we chose different GC cup wheels so as to keep the initial surface to be the same. The workpiece ground is silicon carbide. Table 1 lists the grinding conditions.

3. Experimental results and analysis

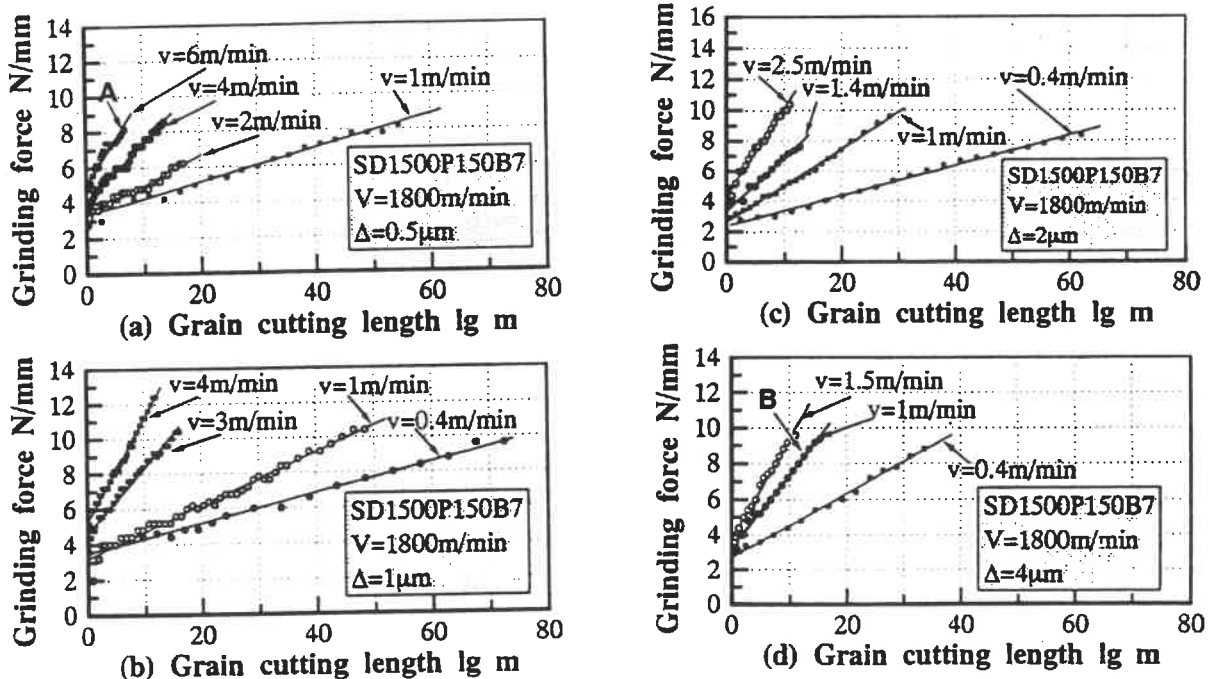


Fig. 2 Grinding forces vs. grain cutting length

Figure 2 (a), (b), (c), (d) show the grinding force variation with cumulative grain cutting length l_g ($l_g = \sum l_c$) when using a resinoid bonded diamond wheel. In the four cases, grinding forces increase linearly with l_c , therefore we can use the slope of the grinding force vs. grain cutting length, called here the grinding force increasing rate (GFIR) Γ , to represent the grinding force variation. Then Figure 2 can be expressed in an other form as shown in Figure 3, which gives the

relationship between Γ and g_m . It is apparent that, when l_c is shorter than 0.5mm, the GFIR is dominated by grain depth of cut g_m . However, when l_c exceeds 0.5mm, both g_m and l_c affect GFIR, and the influence of l_c becomes more significant for longer l_c .

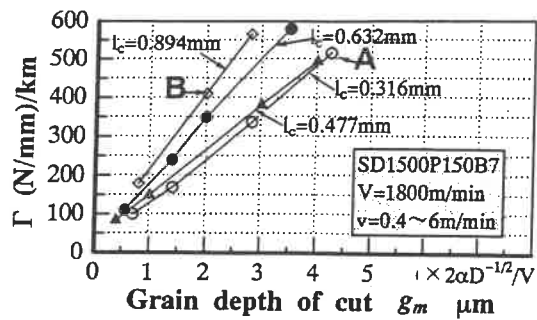


Fig. 3 GFIR of resinoid bonded wheel

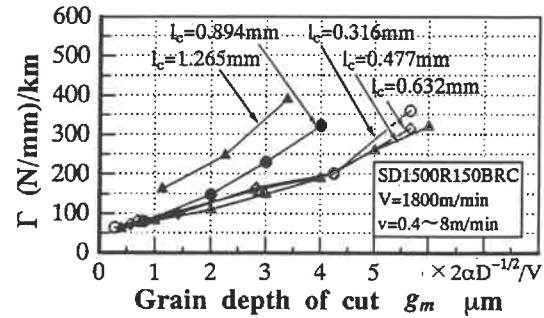


Fig. 4 GFIR of composite bonded wheel

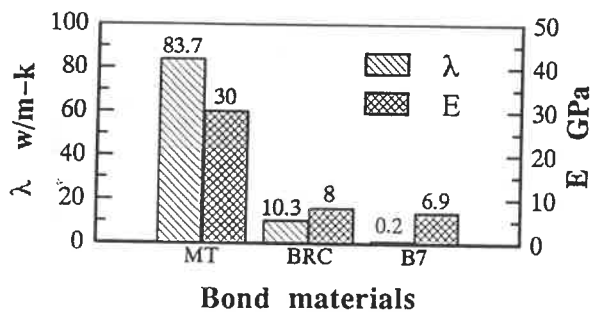


Fig. 5 Thermal conductivity and Young's modulus

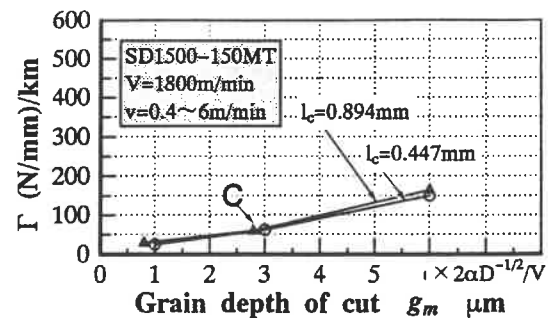


Fig. 6 GFIR of metal bonded wheel

Figure 4 shows the results using a resinoid/metal composite bonded diamond wheel. Here the critical l_c is about 0.6mm, longer than previous. This is thought to be due to the higher thermal conductivity of a composite bond, see Figure 5. Besides, in Figure 4 the slopes of GFIR lines are smaller. We think this is due to the larger Young's modulus of the composite bond. To verify that the thermal conductivity and Young's modulus of bond material have a significant influence on GFIR, we conducted experiments with a metal bonded diamond wheel. Both the thermal conductivity and Young's modulus of the metal bond are much larger, see Figure 5. If they have considerable influence on the grinding results, then when grinding with the metal bonded diamond wheel, the critical l_c should be even longer and the slopes of GFIR line should be even smaller. Figure 6 shows the results when using metal bonded diamond wheel. As can be seen, this is really the case. This time even l_c is substantially longer, it does not affect GFIR, and it seems that the critical length l_c is infinite. Further, the slopes of GFIR lines are even smaller.

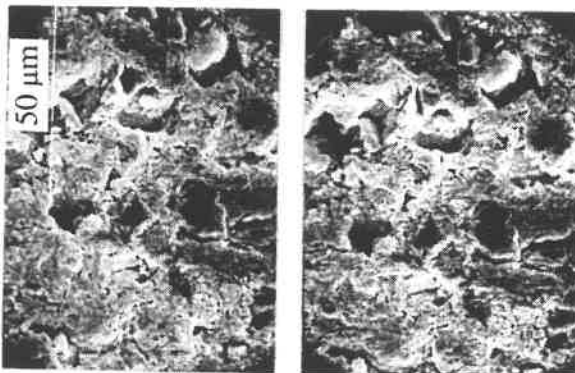


Fig. 7 Elastic deflection of grain

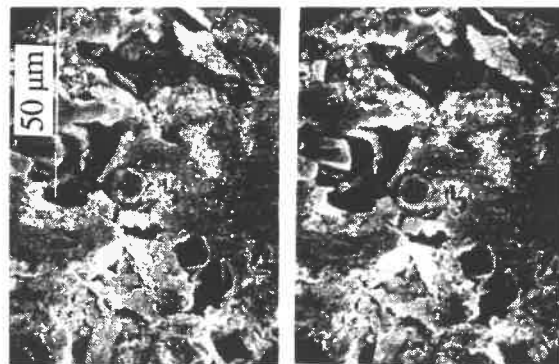


Fig. 8 Plastic sinking of grain

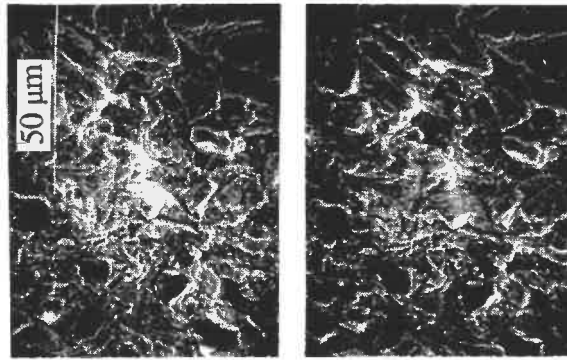


Fig. 9 Attritious wear of grain

For the purpose of understanding how g_m and l_c affect wheel-workpiece interaction in grinding, we observed the wheel surface under two different conditions with the SEM as shown in A and B in Figure 3. The grinding conditions of A and B represent the unique influence of g_m and l_c respectively. Figure 7 is the SEM stereo pairs [2] of a wheel surface topography corresponding to point A. There exist some contact traces on the bond surface even though the measured grain protrusion height only changed in the range of sub microns in 600 grinding passes. This means that in case of larger g_m , elastic deflection of the grain occurs in grinding. Figure 8 shows the wheel surface corresponding to longer l_c , see point B in Figure 3. In addition to the contact traces on the wheel surface, the protrusion height of some grains decreased a several microns in 70 grinding passes, which suggests that not only elastic deflection but also plastic sinking of grains occurred in grinding. Similar phenomena to these were also observed when using a composite bonded wheel. However, when grinding with a metal bonded wheel, even though both l_c and g_m are quite long, attritious grain wear is the dominant wear type. Figure 9 is one of the results, and corresponding to the grinding condition in point C in Figure 6.

Conclusions

Summarizing the results of this investigation, four conclusions were reached:

- (1) There exists a threshold value of wheel-workpiece contact length l_{cth} for a resinoid bonded diamond wheel. When l_c is shorter than l_{cth} , the grinding force increasing rate is smaller and is dominated by grain depth of cut g_m . However, if l_c exceeds l_{cth} both g_m and l_c affect the grinding force increasing rate and the influence of l_c becomes more significant in case of longer l_c .
- (2) The elastic deflection and plastic sinking of grains on a wheel surface were found to be a unique phenomena when using a resinoid bonded diamond wheel with ultra fine grains. Under larger grain depth of cut g_m , elastic deflection of grains occurs. When l_c exceeds l_{cth} , both elastic deformation and plastic sinking of grains occur.
- (3) To insure better grinding performance and longer wheel life, grinding operations employing a diamond wheel with ultra fine grains should be conducted in the range of $l_c < l_{cth}$. That is, for ultra precision grinding of ceramics, a grinding machine with a high resolution feed system is required.
- (4) The Young's modulus and thermal conductivity of bond material have a considerable influence on elastic deflection and plastic sinking of grains respectively. To avoid plastic sinking of grains, which causes a remarkable deterioration in the grinding performances, using bond material with higher thermal conductivity is recommended.

References

1. Abe, K. et al., "Development of ultra precision grinding equipment for ductile mode grinding of brittle materials", Japan Society for Precision Engineering, Vol.59, 12, 1993, 73-78.
2. Zhou, L. et al., "Truing, dressing and grinding characteristics for very fine grit diamond wheels", Proceedings of CIRP Conference on PE & MS, 1991, 233-241.
3. Ichida, Y. et al., "Optimum grinding condition for fine grain diamond wheels in precision grinding of ceramics", Japan Society for Precision Engineering, Vol.58, No. 6, 1992, 145-151.
4. Dornfeld, D. et al., "An investigation of grinding and wheel loading using acoustic emission", Journal of Engineering for Industry, vol. 106, No. 1, 1984, 28-33.

PART SPEED EFFECTS ON SURFACE ROUGHNESS IN PRECISION GRINDING

Gene E. Storz, Thomas A. Dow
Precision Engineering Center
North Carolina State University

INTRODUCTION

To understand the mechanisms of contour grinding, it is necessary to understand the geometry of the process producing both the surface roughness and subsurface damage. Over the past few years at the Precision Engineering Center, the effects of the geometry have been studied and developed into a model called the Virtual Contour Grinding Model[1]. The model combines the effects of wheel speed (N_w), part speed (N_p), wheel radius (R), nose radius (A), crossfeed rate (cf), and part diameter (D) to create a theoretical surface and grinding chip. Several values can then be calculated for these objects such as surface roughness, chip thickness, damage generated, etc. The model is limited to combining only the geometry effects of grinding (Boolean geometry) using the assumption that a single region of the wheel performs the majority of cutting once per revolution. Although the model is limited in this aspect, it does show the interaction of the base geometry and give predictions for grinding conditions to obtain better surface characteristics. The Virtual Contour Grinding Model was also used to generate data from which theoretical curves for surface roughness and chip thickness were developed. The current focus of research is to verify these models and determine how they can be used in application. The part speed has been found to be of particular importance to surface roughness and will be discussed in the following sections.

SURFACE ROUGHNESS AND PATTERN INDEX

Using the Virtual Contour Grinding Model, a set of theoretical surface roughness curves were developed using the theoretical maximum surface roughness and the pattern index. The pattern index is an analytical expression developed from relationships in the surface roughness data calculated from the Virtual Grinding Model under varying grinding conditions. The maximum surface roughness, RMS_{max} , is a theoretical equation derived by Fawcett[2] using the geometry of contour grinding. The pattern index, defined as C in Figure 1, and the maximum surface roughness can be calculated using Equations (1) and (2). Since the effective cutting diameter changes during the grinding operation, an average diameter is used for all calculations.

The pattern index determines the shape of the surface roughness curve according to the legend shown in Figure 1. The maximum surface roughness is then multiplied by the point on the

characteristic curve at the appropriate phase to yield an estimate of the geometric surface roughness. The phase (ph) is determined by the ratio of wheel speed to part speed given by Equation (3). The INT specifies the integer portion of the calculated value. The upfeed (uf) and crossfeed (xf) are given by Equations (4) and (5).

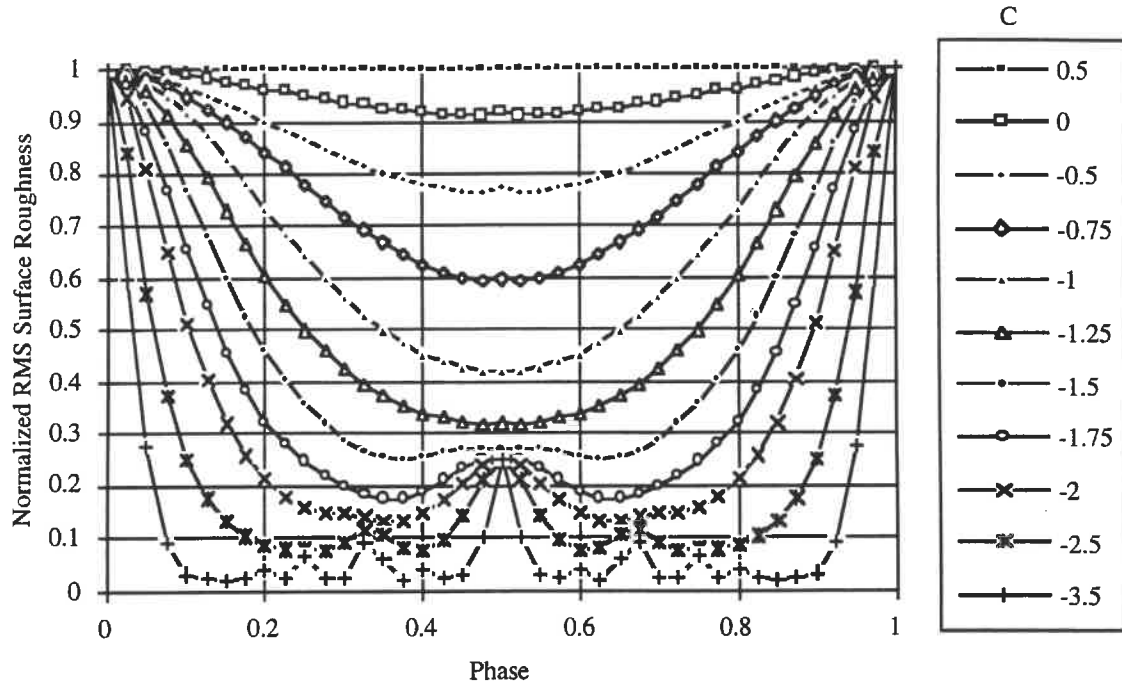


Figure 1: Theoretical surface roughness curves for contour grinding

$$C = \log \frac{R}{A} - 2 * \log \frac{uf}{xf} \quad (1)$$

$$RMS_{max} = \sqrt{\frac{uf^4}{720 * R^2} + \frac{xf^4}{720 * A^2}} ; mm \quad (2)$$

$$ph = \frac{N_w}{N_p} - INT \frac{N_w}{N_p} ; radians \quad (3)$$

$$uf = \frac{\pi * N_p * D}{N_w} ; mm \quad (4)$$

$$xf = \frac{cf}{N_p} ; mm \quad (5)$$

The pattern index and maximum surface roughness were developed separately and seemed to have no relationship. However, careful examination of the maximum surface roughness over a range of

part speeds revealed a minimum at a pattern index value of 0. The maximum surface roughness and the pattern index were explored more closely to determine if a direct relationship existed.

If a minimum RMS_{max} value occurs over a range of part speeds at a pattern index value of 0, then a relationship must exist between the pattern index and the derivative of the maximum surface roughness. First, the upfeed and crossfeed are substituted into Equations (1) and (2) so that both equations are a function of part speed. Next, the pattern index (Equation (1)) and the derivative of Equation (2) with respect to part speed are set equal to 0. Solving both equations for part speed yields Equation (6), a direct relationship between the maximum surface roughness and the pattern index.

$$N_p = \left(\frac{cf * N_w}{\pi * D} \right)^{\frac{1}{2}} * \left(\frac{R}{A} \right)^{\frac{1}{4}} ; rpm \quad (6)$$

Not only does Equation (6) predict at what part speed the minimum RMS_{max} value will occur, it also predicts when variations in the surface roughness will begin to occur due to phase. Figure 1 shows that a pattern index value greater than 0 will maintain the maximum surface roughness independent of the wheel and part speeds. As the pattern index approaches 0, the surface roughness curves begin to vary with phase. As the pattern index drops below 0, the variation of surface roughness with phase continues to increase indicating that large variations in surface roughness will occur on a surface with a low pattern index value. To predict a lower bound for the surface roughness, RMS_{min} , the normalized minimums in Figure 1 were examined as a function of the pattern index, C , resulting in Equation (7).

$$RMS_{min} = 10^{(-0.0348 + 0.17855 * C - 0.165 * C^2 - 0.02374 * C^3)} * RMS_{max} \quad (7)$$

EXPERIMENTAL VERIFICATION

To verify the theoretical results, experiments were conducted using the Small and Large Form Grinders at the PEC. The results obtained from the Small Form Grinder are shown in Figure 2. The theoretical maximum and minimum surface roughness given in Equations (2) and (7) are plotted as a function of the part speed. The minimum surface roughness starts at a part speed of approximately 50 rpm as calculated from Equation (6) where phasing begins to occur. As the part speed increases beyond 50 rpm, the theoretical maximum and minimum surface roughness continue to diverge. This indicates that both the pattern index is becoming more negative and the surface roughness as measured on a ground surface will have larger variations.

Experimental tests were conducted at part speeds of 5, 20, 50, 75, 125, and 250 rpm. Surface roughness measurements were then made on a Zygo Mark IV Interferometer. At low part speeds, the variation in surface roughness was small. At 50 rpm, a variation in surface roughness was

detected at different locations on the surface. These variations continued to increase as the part speed was increased as shown in Figure 2. Experiments on the Large Form Grinder show similar results.

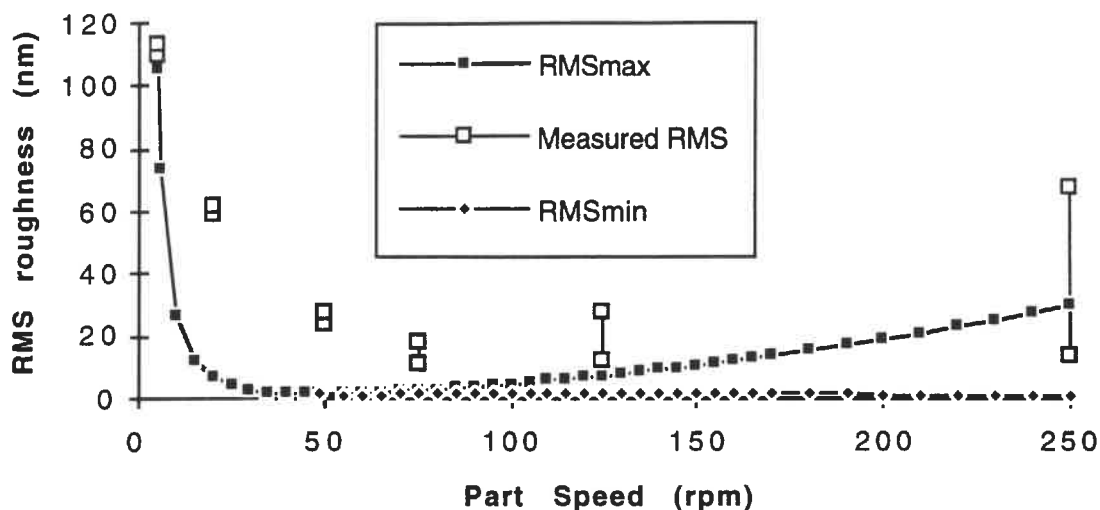


Figure 2: Theoretical and experimental RMS surface roughness for $R = 7$ mm, $A = 5$ mm, $D = 6$ mm, $cf = 0.596$ mm/min, $N_w = 63000$ rpm. Material = CVD Silicon Carbide

CONCLUSION

Both the model and the experimental measurements indicate that an optimum part speed can be chosen to obtain a minimum surface roughness for given grinding conditions. The experiments also showed that a minimum surface roughness is reached soon after phasing begins to take place and the variation in the surface roughness increases with increased part speed. Therefore, it is desirable to use part speeds slightly above those calculated by Equation (6) to obtain a minimum surface roughness with minimum variation. A lower part speed is also conducive to minimum damage generation[1].

REFERENCES

- [1] Storz, G.E., "Theoretical Modeling of Contour Grinding", MS Thesis, North Carolina State University (1994).
- [2] Fawcett, S.C., "Development and Implementation of a Grinding Technique for Precision Finishing of Brittle Materials," Ph.D. Dissertation, North Carolina State University, 1991.

GRINDING TEMPERATURE IN CERAMIC WORKPIECES

Shao-xiong Liang, Owen F. Devereux and Ya Yun Li

University of Connecticut

Storrs, CT

Grinding of ceramics is commonly accomplished with a rigid grinding machine using a diamond wheel. Due to the brittleness and hardness of ceramic workpieces, grinding inevitably induces damage to the workpiece. As in metal grinding, where thermal damage is a major concern and has been extensively studied [1], high surface temperature and temperature gradients may also be the primary cause of grinding damage in ceramic materials. (In a simple experiment a ZrO_2 workpiece cracked when heated locally to $\sim 400^\circ C$ by a moving diode laser source.) A high temperature gradient over space and time during grinding can induce residual stress, phase transformation and microcracking in the workpieces. The intense heat of grinding and the thermal cycles applied to the diamond wheel can also degrade the bond strength within the abrasive layer, weaken the fracture strength and reduce the wear resistance of diamond grits, leading to a shorter wheel life [2]. Therefore, temperature measurement and prediction are very important to the ceramic grinding industry.

Usually, temperature is measured by thermocouples, as this technique is easy and low in cost [3]. The primary limitations of this technique as applied to localized, transient measurement are the response time and spatial resolution. Due to the steep temperature gradients near the grinding surface with respect to both the time and space, experimental estimation of grinding temperature at the surface is difficult.

To enhance the response speed, K type thermocouples of 0.0005" and 0.001" diameter were used in our experiments, producing response time of less than 4 ms in blowing air, whereas the shortest time for the grinding interface to pass a thermocouple was ~ 40 ms for the grinding conditions employed. To ensure tight contact between the thermocouples and the ceramic workpiece, the thermocouples were "sandwiched" between two ceramic workpieces bonded with a thermally conductive paste applied to the interface, as shown in Fig.1. Grinding temperature and forces were measured under the different grinding conditions, at various table feed rates and depths of cut, and the temperature vs. depth profile was determined for dry grinding.

Temperature and its distribution during grinding were also studied by Finite Element Modeling(FEM) using MARC[®] software. The generation of heat during the grinding process was simulated by a moving heat source passed over the workpiece. The heat created at the grinding arc is transferred to the workpiece, wheel and coolant and, to simulate the temperature distribution, a heat partitioning factor, R , the fraction of the heat generated that enters the

workpiece, had to be estimated assuming one dimensional heat transfer and equality of temperatures of the workpiece, wheel and the coolant at the grinding interface:

$$R = \frac{1}{1 + \frac{v_w}{\lambda} \sqrt{\frac{(k\rho c_p)_{wh}}{(k\rho c_p)_{wk}}} \cdot C \cdot \pi \cdot d_g \cdot L \cdot \sqrt{\frac{a}{D_s}} + \frac{v_w}{\lambda} \sqrt{\frac{(k\rho c_p)_{col}}{(k\rho c_p)_{wk}}} \cdot \left(\frac{v_s}{v_w} - C \cdot \pi \cdot d_g \cdot L \cdot \sqrt{\frac{a}{D_s}}\right)} \quad (1)$$

In equation (1) k , ρ , c are the thermal conductivity, density and the specific heat of the materials, respectively, and the subscripts wh , wk and col represent the wheel, workpiece and coolant. C is the surface density of the wheel abrasives, d_g is the average grit diameter, L is the average distance between two effective abrasive grits on the wheel surface, a is the depth of cut, v_w is the table speed, v_s is wheel peripheral speed and λ is a disposable parameter that can be determined experimentally.

The results of this study are shown in Figures 2 through 5. Fig.2 shows the variation of measured near-surface temperature with material removal rate for two different table speeds. Grinding temperature increases with material removal rate and is smaller for higher table speeds. The effectiveness of coolant in reducing grinding temperature is shown in Fig.3. Obviously, dry grinding produces much higher temperatures than wet grinding. Fig.4 shows the temperature vs. depth profile for dry grinding; the results of measurement and FE modeling are compared. It can be seen that FE modeling is consistent with the experimental measurement, and that the temperature gradient is steep near the surface. Fig.5 compares the results of modeling and measurement at different material removal rates. Again, FE modeling successfully predicts the grinding temperature and its variation with material removal rate.

In conclusion, near-surface temperature measurement in ceramic grinding by thermocouples is accurate, cheap and convenient. In general, grinding temperature increases with increasing material removal rate. Temperature is higher during creep feed grinding and smaller at higher table speeds for a fixed material removal rate. Obviously, coolant application is very effective in reducing grinding temperature. The temperature vs. depth profile obtained indicates that the temperature gradient is steep within 50 μm of the surface of the workpiece. Temperature distributions predicted by FEM are consistent with experimental observation. We believe that this modeling tool will be useful for predicting grinding temperature distribution, which in turn is important for determining thermal stress and phase transformation in the workpiece and thus for optimization of ceramic grinding parameters.

References

1. Malkin S. and Anderson R.B., "Thermal Aspect of Grinding, Part 2-Surface Aspects of Grinding," Transactions of ASME, Nov. 1974, pp.1184-1191.
2. Jahanmir S., et al, "Ceramic Machining: Assessment of Current Practice and Research Needs in the United States," NIST Special Publication 834 (1992), pp. 51-63.
3. Yokomizo S. and Kaneeda T., "Fine Ceramics Grinding Temperature," Proceedings of ASPE 8th Annual Meeting, Nov. 7-12, 1993, Seattle, Washington, pp 221-224.

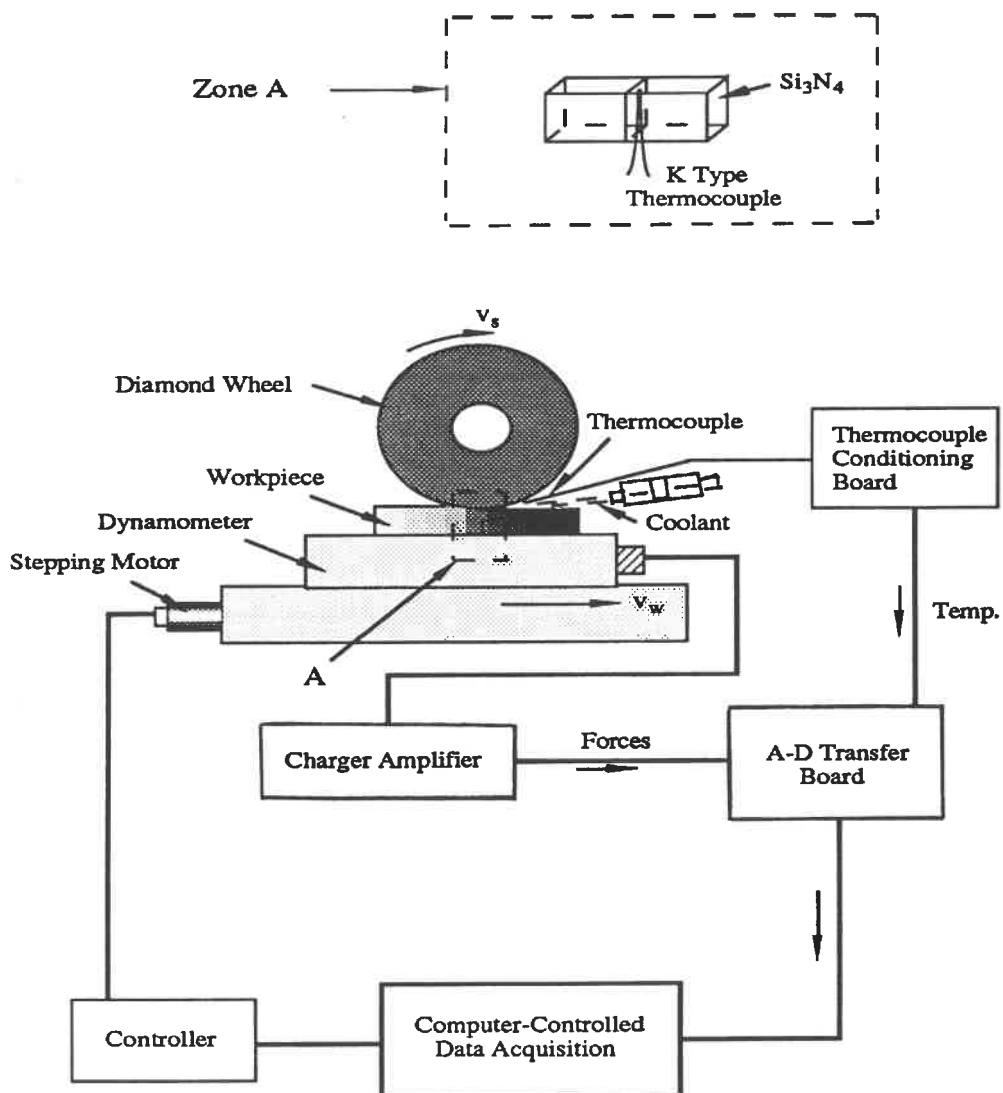


Fig.1 Schematic of experimental setup.

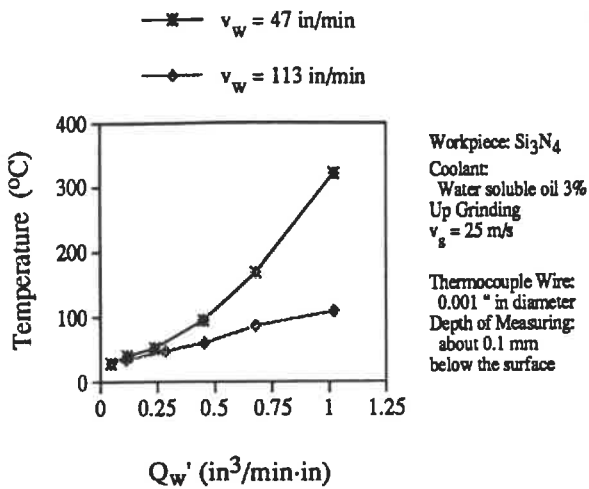


Fig.2 Grinding temperature vs. MRR for two table speeds.

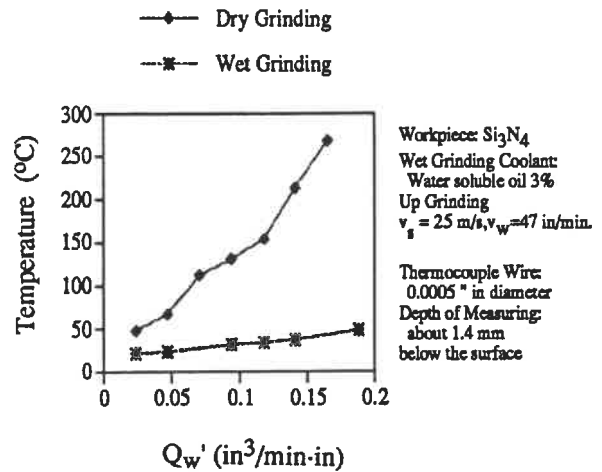


Fig.3 Temperature vs. MRR for dry and wet grinding.

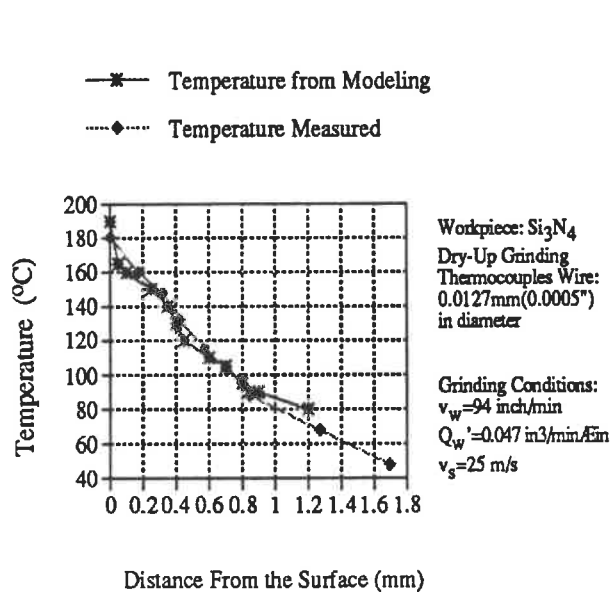


Fig.4 Temperature vs. depth profiles obtained from modeling and experiment.

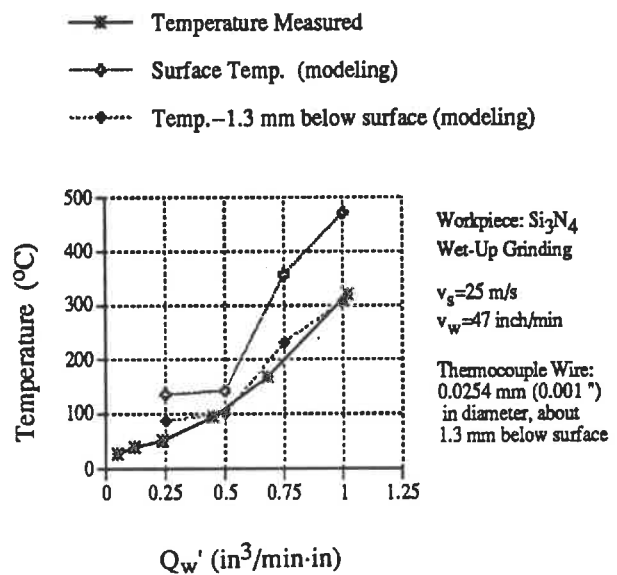


Fig.5 Comparison of grinding temperatures from modeling and measurement.

IMPLEMENTATION OF A REAL-TIME GRINDING PROCESS ESTIMATOR

Hodge E. Jenkins

Georgia Institute of Technology
The George W. Woodruff School
of Mechanical Engineering
Atlanta, GA 30332-0405

Thomas R. Kurfess

Georgia Institute of Technology
The George W. Woodruff School
of Mechanical Engineering
Atlanta, GA 30332-0405

Abstract

Precision components require an ever increasing quality of surface finish as well as a reduction in processing time. Thus, it is desirable to control the grinding process in terms of position, velocity and force. Existing fixed-gain control systems are limited because of process variation. Model reference adaptive control can improve these limitations. To accomplish this a time varying, parametric model relating the material removal rate (MRR) to grinding forces as well as other process variables, is needed to describe the grinding process. Using a time-varying model as a basis, this paper develops real-time estimation techniques for grinding applications. The real-time model is implemented as part of an adaptive control system to maintain a desired MRR and limit force with changing process parameters.

Previous research indicates that several measurable sources of data can be correlated to grinding model parameters. Analysis of multiple sensor data using Kalman filtering and other multi-variable recursive techniques has achieved a superior grinding model parameter estimation, suitable for adaptive control of force and displacement trajectories. This paper develops multi-sensor input and windowing techniques for estimation of the process model parameters. Simulations, using controlled grinding force experimental data, compare the real-time parameter estimation model results to actual data. Implementation issues for real-time systems are also discussed.

Grinding Process Modeling

To improve the quality of a ground surface it is desirable to control the grinding process in terms of positional accuracy, uniformity of velocity and force limits. The grinding process varies continuously with changing properties and conditions in the grinding wheel, material, and existing surface profile. Therefore, a real-time grinding process model is needed to provide a robust controller suitable for creating precision surfaces, in a timely manner. An observer (or estimator) is useful for identifying process parameters to be used in adaptive force and position controllers that achieve a desired profile and surface finish with minimal damage.

Before an adaptive control scheme can be implemented, an appropriate form for the model must be selected. The combined grinding model of Hahn and Lindsay (1971) and Preston (Brown 1990) relates power (the product of force and velocity) to the MRR (equation 1).

$$Z = K_p (F_N - F_{TH}) V; \quad \forall F_N \geq F_{TH} \quad (1)$$

This equation has been previously used to define the relationship between the normal force, wheel contact speed, and the MRR (Brown 1990). Z represents the material removal rate, while F_N and F_{TH} are the normal force and threshold normal force, respectively. The threshold value of the normal force is the value below which insignificant material removal takes place. The coefficient, K_p , correlates the normal grinding force to the expected removal rates, given the specific conditions of the workpiece and

wheel. V is the wheel surface speed. Previous work indicates the grinding model of equation (1) can adequately represent the grinding process over short durations of time, with constant coefficients. It has also been shown that the threshold force, F_{TH} , remains relatively constant (Ludwick et al., 1994). Therefore, this work only concentrates on estimation of K_p .

Parameter Estimation

It has been shown that the MRR is correlated to several sensor values, including tangential cutting forces as well as normal forces in plunge grinding. There is a high correlation of the average normal to tangential grinding force ratios to MRR. The higher ratios signal the presence of a "rubbing" condition. Other correlations should exist with other sensory data such as wheel speed or input power. Thus, it is proposed that additional sensors may be employed to extract and utilize additional information from each sensor for improved estimation.

Several methods are implemented and evaluated for estimating the grinding model parameters. To remove noise and random machine vibrations from the model parameter estimation, traditional low-pass filtering of the sensor data is applied. This provides adequate filtering for parameter estimation results in many applications. However, in real-time applications of control and prediction, recursive methods are generally needed for integrating force sensor, velocity, and power data as inputs for grinding process parameter estimation. For using multiple sensor information, recursive methods are limited to general recursive least squares (RLS) forms such as RLS with a forgetting factor (RLS-FF), gradient approaches, and the Kalman filter.

Figure 1 outlines a windowing procedure to be used in an adaptive parameter estimation and control framework for a Kalman filter. The sensor variance of the data in a previous window is used to update the Kalman gain in the following window. The sequence begins with a window for calculating variance. These statistics are used by a subsequent window for estimating parameters. The identified parameter may then be used to determine the controller gain. Successful application of this approach must resolve any Kalman filter design issues, including stability, accuracy and timeliness of noise covariance estimation, speed of the filter, and the rate of parameter variation.

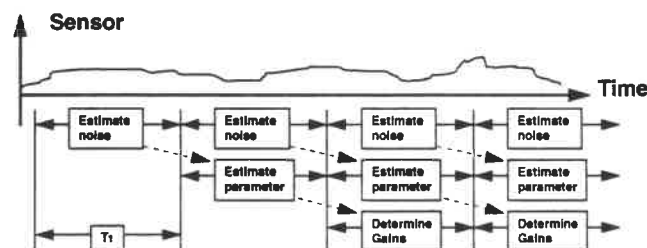


Figure 1. Windowing of Sensor Data for Variable Parameter Estimation and Adaptive Control.

Results

MATLAB procedures were used to develop several algorithms for estimating K_p , and simulating material removal. Specifically four techniques were examined (low-pass filtering, windowed ARX, RLS-FF, and windowed Kalman filtering). Experimental grinding data (normal force, tangential force, and tool end displacements) were used as input to the simulation (Ludwick et al,

1994). The real-time grinding model coefficient, K_p of equation (1), was calculated based on the recorded sensor data. The error variance between the actual and the simulated displacements is used for the numerical comparison of the different estimators. Displacement error variances are summarized in Table 1. The displacement error variances are summed for the first 1,000 data points (4.5 s).

Figure 2 (a) depicts the determined real-time grinding model coefficient, K_p , and its average value, during a force-regulated grinding experiment with 4.5 ms sampling time. The real-time values of K_p were calculated using equation (1), and low-pass filtering at 10 Hz to reduce noise from the numerical differentiation. Clearly, the variation of K_p is evident; indicating that a constant value is not appropriate for more precise grinding estimation. Figure 2(b) plots the simulated and actual tool displacements. The lower line represents the simulated tool displacements using the real-time estimates of K_p , while the upper line is the actual measured tool displacement. As K_p is always positive or zero, the simulated tool displacement curve represents the ground part surface.

Figures 3(a), (b), and (c) plot the simulated and actual tool displacements for a windowed Kalman estimator (20 point window), an ARX technique (50 point window), and a RLS-FF method. All the recursive methods used in these simulations performed well, and improved the estimate of K_p .

Window size variation was examined for the effect on the displacement error covariance. From our data there appears to be an optimum window. The window size represents a trade-off between accuracy and timeliness. There is also an optimum window for the ARX approach. Akin to the optimum window size, there is an optimum forgetting factor of $\lambda=0.8$ for the RLS-FF method.

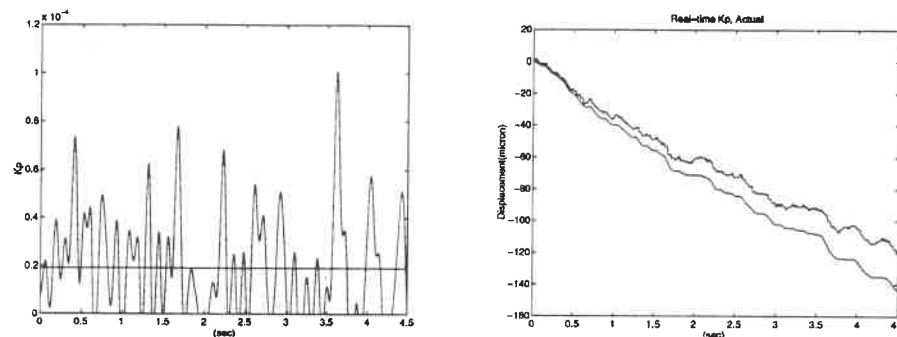
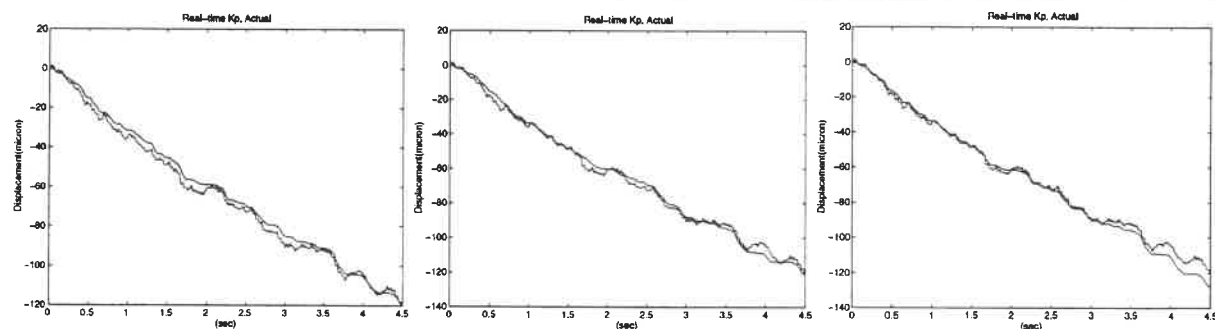


Figure 2 (a). Real-Time Preston Coefficient (b). Simulated and Actual Displacements.



(a). Windowed Kalman Filtered (b). Windowed ARX (c). RLS-FF, $\lambda=0.9$

Figure 3. Simulated and Actual Displacements

Table 1. Displacement Error Variance.

Selected Case	Error Variance
Real-time K_p Low Pass Filtered 10 Hz	161
Kalman Filter, 10 pt. window	234
Kalman Filter, 20 pt. window	10
Kalman Filter, 50 pt. window	74
R-T K_p , ARX, order 2, 100 pt. window	41
R-T K_p , ARX, order 2, 50 pt. window	7
R-T K_p , ARX, order 2, 20 pt. window	76
RLS-FF, order 2, $\lambda=0.9$	17
RLS-FF, order 2, $\lambda=0.8$	11
RLS-FF, order 2, $\lambda=0.7$	15

Conclusions

Multi-sensor integration methodologies for estimation of a real-time grinding model coefficient have been successfully simulated, using actual experimental grinding data. Variation of the model's coefficient appears to account for differences between previous constant coefficient model simulations and actual tool displacements. Several recursive techniques including a windowed Kalman filter, a windowed ARX estimation, and RLS-FF techniques have yielded excellent improvements in estimating the grinding model coefficients over conventional low-pass filtering alone.

The best result obtained for the each of the applied parameter estimation techniques shows an order of magnitude reduction in the error covariance, as compared with low-pass filtering alone. Selection for implementation of a windowed Kalman filter, a window ARX estimation, or a RLS-FF will depend on the robustness of the algorithm and its time efficiency.

For the examined techniques, the window size and model order have a strong influence on the accuracy of the result. The optimum window size represents a trade-off between the timeliness of the data and the reduction of the noise covariance. Additional sensor inputs may be able to reduce the error somewhat further and should be investigated. These results should translate into improved controllers and a potential on-line diagnostic tool for grinding.

References

- Brown, N., 1990, *Optical Fabrication*, MISC4476, Revision 1, Lawrence Livermore Laboratory, Livermore, CA.
- Danai, K. and Ulsoy, A. G., 1987, "An Adaptive Observer for On-Line Tool Wear Estimation in Turning, Part I," *Mechanical Systems and Signal Processing*, Vol. 1, No. 2, pp. 211-240.
- Hahn, R. S. and Lindsay, R. P., 1971, "Principles of Grinding Part I Basic Relationships in Precision Machining," *Machinery*, July, pp. 55-62.
- Ljung, L., 1987, *System Identification: Theory for the User*, Prentice Hall, Englewood Cliffs, NJ.
- Ludwick, S. J., Jenkins, H. E., Kurfess, T. R., 1994, "Determination of a Dynamic Grinding Model," *1994 International Mechanical Engineering Congress and Exposition*, DSC-Vol. 55-2, pp. 843-849.
- Ulsoy, A. G., 1989, "Estimation of Time Varying Parameters in Discrete Time Dynamic Systems: A Tool Wear Estimation Example," *1989 American Control Conference*, Vol. 1, pp. 68-74.

IN PROCESS MONITORING AND CONTROL OF INTERNAL GRINDING IN HIGH STRENGTH STEEL

Ma Ping ,Wang Min

Department of Mechanical Engineering, Nanjing University of Aeronautics & Astronautics, Nanjing ,
China P.O. Box 210016

ABSTRACT

The quality of internal grinding is influenced by many factors such as the grinding wheel characteristics , dressing conditions and grinding parameters. In this paper, a new advanced technology, internal grinding in high strength steel with Cubic Boron Nitride (CBN) wheel , is introduced as well as a special set-up is developed based on the general machine tool. According to the investigation of the characteristics of process, three steps have been taken to ensure the process and improve the grinding quality. First is the optimizing of the dressing condition, in order to obtain good grinding ratio , the protruding height and chip space ratio must be modified in proper condition . Second is the improvement of rigidity of the mechanical structure, this paper investigates the features of the parts, and a new structure is developed . The third is the establishment of the adaptive controller based on the normal and tangential force. Compared with the general external grinding process, the internal grinding based on the special set-up lacks good rigidity. In order to reduce its influence on the grinding efficiency, the variation of the grinding force is studied comprehensively and an effective monitoring strategy is introduced while corresponding adaptive controller is built up. The tests show that the grinding force is most effective variable to detect the internal grinding condition, including the variation of the actual grinding depth and grinding burn.

key words: Internal grinding , CBN wheel, grinding force, in-process monitoring.

1. Introduction

300M is a high strength steel with excellent toughness and difficult to machine, widely applied in Aeronautical and Astronautical industry. By heat treated, its hardness reaches nearly HRC 50 ~ 55. In this paper, an advanced technology, the internal grinding of deep hole in 300M , is discussed. Studies show that the conventional wheels such as corundum wheel, cannot meet the requirements because of its poor wear resistance , lower grinding ratio and shorter dressing interval. Those wheels keep sharp in short time, furthermore, the integrity and profile of the product cannot satisfy the demands , especially it is easy to produce cone along the deep hole. In this work, the Resin Bonded Cubic Boron Nitride(CBN)wheel is introduced and the characteristics of this process are investigated.

As we know, the CBN wheel possesses excellent wear resistance, when it is well dressed, its dress interval is 50 to 100 times than general wheel. So high grinding ratio and the precision of profile can be achieved with CBN wheel.

300M is a low alloy high strength steel with low tempering temperature. its mechanical property is sensitive to surface defect, if the thermal damage exists , its anti-fatigue life shortens sharply.

To prevent grinding burn, this paper proposes two effective methods, including the improvement of wheel dress and the establishment of monitoring and control system based on the grinding force.

Wheel dress set-up

We know that the binder of CBN wheel is made of phenol resin with good elasticity, low ignition and excellent grains maintenance, where CBN abrasives are covered in the resin, leading to the high density texture and few air holes. To improve its grinding capability , CBN wheel needs to be dressed properly. Previous works have indicated that two major factors relative to grinding capability are the protruding height and chip space ratio[1][2]. Traditionally, the Resin Bonded CBN wheel is dressed

by the single point diamond dresser. This kind of dressing method is simple and conventional, moreover, the dressing force merely produce derivation of the spindle, which is helpful to obtain proper wheel shape. However the chip space ratio is poor and vast dress would speed up the wear of the wheel.

In this paper, we develop a new dress method, its procedures are drawn as following:

The primary dress is carried out by the SiC wheel to obtain the basic shape as fig1.a. The single point diamond pen follows to refine the wheel surface shown in fig1.b. The third step is to sharpen the wheel with the elastic bend while the fluid mixed with oil and SiC abrasives are sprayed into the contacting zone illustrated as figure1.c. The corresponding variation of the wheel surface is changed as figure 2.

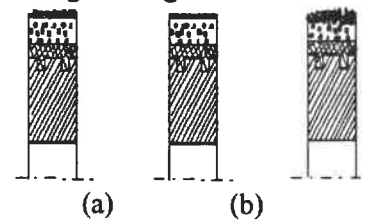
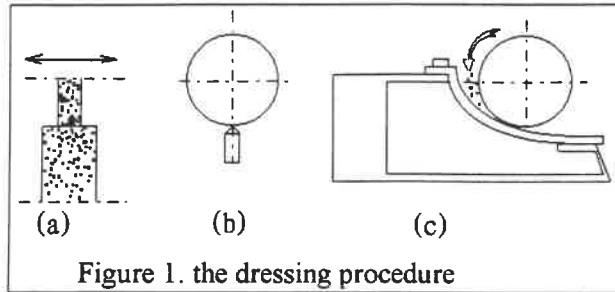


Figure 2. The wheel varies in (a) SiC dressing (b) diamond dressing (c) elastic bend dressing

In order to optimize the dress condition, a series of tests have been carried out and the protruding height of grains and the chip space ratio are investigated systematically.

The experimental conditions:

Machine tool: CW61100E. Feedrate: 1.5um/rev. Depth of cut: 10um.

Wheel: Resin Bonded CBN wheel $P75 \times 20 \times 32$ CBN100[#] / 120[#], $V_s=30$ m/s.

The influence of the protruding height of grain and the chip space ratio to the grinding ratio are shown as figure 3.

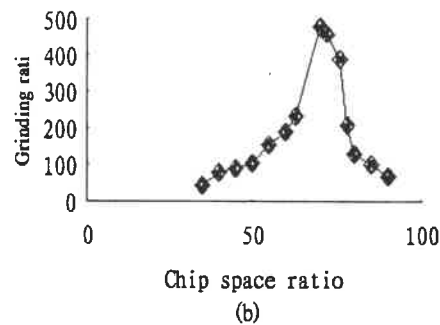
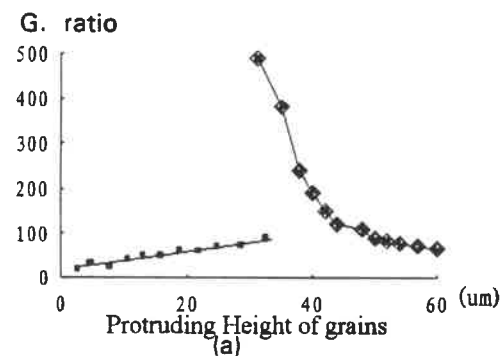


Fig.3. The variation of grinding ratio with difference of

protruding height and chip space ratio

Two regions can be identified in figure 3.a, where the statistics of the protruding height of grains are

$$\text{derived as: } h = \frac{1}{n} \sum_{i=1}^n h_i \quad (1)$$

According to figure 3.a, when h is less than $3.10a_p$, the grinding ratio is lower than 100. On the other hand, when h exceeds the limit, the variation of grinding ratio approximates an exponent curve,

expressed as $G=ah^b$ ($b<0$), the expression can be drawn by gressive analysis as equ(2):
 $G = 0.396 \cdot 10^{-3} h^{-4.21}$, (2)

Studies show that when the grinding ratio is much small, the grinding burn takes place easily because thermal decomposition temperature of the binder is about 300°C to 400°C.

The other major factor influence to the grinding ratio is chip space ratio. According to fig3.b, Grinding ratio varies just like a probability curve, which could be described as $y=a \exp(-k_c^2)$. The empirical formula is also obtained: $G=485.463\exp[-0.0175(K_c-78.6)^2]$ (3).

Tests highlight that When the h is less than the limit($3.10\alpha_p$), and the K_c is more than 92 or less than 55, it easily expedites the friction and rubbing action between wheel and workpiece, provoking the wheel blocking and increasing the grinding temperature simultaneously. As the temperature goes beyond the thermal limit(300 °C ~ 400 °C) the carbonization of binder would accelerate the grits dropping, then grinding ratio would decrease exponentially.

Proper wheel dress is much more important. Synthesizing those factors, when h is larger than 3.01 then expression of grinding ratio can be described as

$$G=\min\{0.396 \times 10^{-3}h^{-4.21}, 485.463\exp[-0.0175(k_c-78.61)^2]\} \quad (4)$$

During the refining process, when the spraying speed of mixture of 120[#] SiC abrasive and oil is at speed of 1.5m/min, infeed velocity is 0.16mm/rev, depth of cut 0.01mm, and sharpen time keeps within 4 to 6 minutes, the proper protruding height and chip space ratio could be achieved.

Experimental System

The aeronautical parts is much complicated with 890mm long and its diameter is 80mm. In order to smooth grinding process, a special attachment is developed to fit the conventional machine tool shown as fig.4.

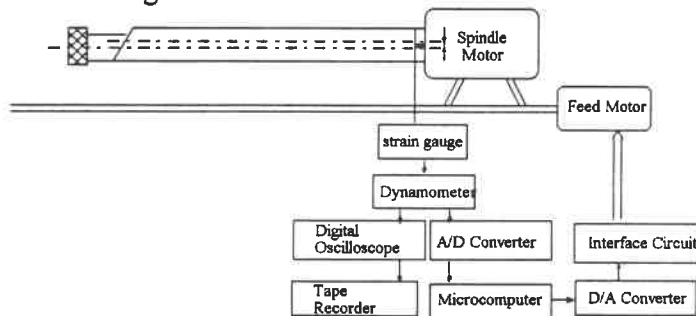


Fig 4. The experimental setup

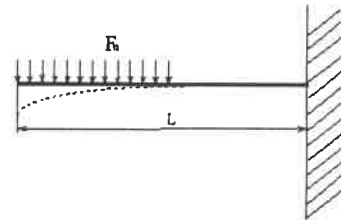
Fig 5. the explicit kinematics of the spindle

On basis of fig.4, the grinding spindle works like a cantilever beam. The eccentric spindle construction has been introduced to improve its rigidity. It consists of two semicircles with radius 37mm. Deviation is designed as 4mm, which reinforces the static stiffness of mechanics greatly. In order to investigate the difference between the nominal grinding depth and the actual grinding depth, the elastic deformation in mechanical structure caused by the grinding force is analyzed as fig.5. The equations are drawn as:

$$\delta = -\frac{F_n}{E}, \quad (5)$$

$$E = \frac{24KJ}{b^3 - bl^2b + 8l^3}, \quad J = \frac{\pi d_s^4}{64}$$

Where δ is the value of deflection and E the static stiffness of mechanical structure, F_n the radial grinding force. b is the grinding width, l the spindle length, J the modulus of elasticity to spindle and d_s the spindle diameter.



During the grinding process, the deflection varies with the change of surface state. For the chrome corundum wheel, as it goes blunt in short time, number of worn or loaded grits increases the grinding force quickly, especially the friction and rubbing component, resulting in the increasing of the deflection. On the contrary, the grinding force of the CBN wheel is rather small, mainly consists of cutting component, and it achieves excellent wear resistance. The deflection changes relatively slightly during the internal grinding process. A series of experiments have conducted on the developed machine tool, and results are obtained as fig.6.

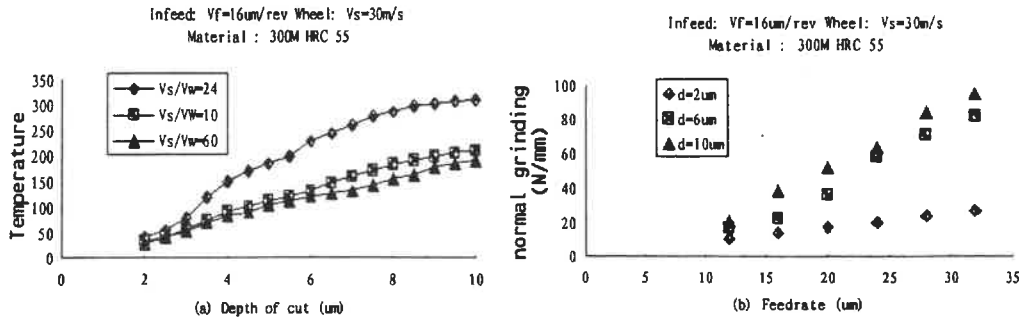


fig.6 (a) the variation of temperature with the change of depth of cut

(b) the relationship between the infeedrate with the normal force

According to fig.6.a, the grinding temperature increases with the rise of velocity ratio. In this process, the speed ratio V_s/V_w is 60, and the depth cut does not exceed 40μm.

Based on fig.6.b, the normal grinding force varies with the change of the feedrate linearly to prevent the grinding burn. The adaptive control of system is established as fig.4, which is equipped with both a spindle and infeed drivers, and an open loop controlled, stepping motor driven table. The programmed feedrate could be overridden with a voltage from the interface circuit connected with control microcomputer, in which 8 bit digital to analog converter is utilized.

The two components as radial and tangential direction force are measured by the strain gauge. The signals from the dynamometer are amplified firstly. Then followed by a low-pass (300 Hz) filter, in the end signals are sampled by 8 bit analog-to-digital converter. The digital signal is transmitted to the control computer.

Grinding process modeling and control

The studies show that the grinding force is a series of stochastic signals [3][4]. During the normal state, the signals are rather stable, but when the wheel dull or overcut, the signals vary obviously. In order to realize the constant force grinding, the time series auto regressive model is introduced.

$\{x_t\}, t=1, 2, \dots, N$, a set of dispersing signals, $x_t = F_n - F_t, F_n$ the radial force, F_t the tangent force.

$$\text{The difference equation is } \hat{x}_t = \sum_{i=1}^n \phi_i x_{t-i} - a_t \quad (6)$$

Where ϕ_i is the i th parameter of the model. n the order of the modes. a_t is residual, obeying the Gaussian distribution, $a_t \sim NID(0, \sigma_a^2)$, σ_a^2 , the variance of $\{a_t\}$.

The parameters of the model can be estimated by levson algorithm.

$$AIC(n) = N \ln \sigma_a^2 + 2n \quad (7), \text{ according to it, the order } n \text{ is obtained.}$$

Test condition: The velocity of wheel V_s is 30m/s, that of workpiece V_n is 30m/min, infeedrate 16um/rev, depth of cut 10um, sample frequency 1.2k, other conditions are similar to the above, the AR model can be expressed as:

$$\hat{x}_t = 0.63x_{t-1} - 0.30x_{t-2} + 0.14x_{t-3} + 0.08x_{t-4} + \alpha_t \quad (8)$$

The square sum of residual $\sigma^2=20.8$, $AIC(4)=1290.7$. Both the estimation $\hat{x}_t$ and the measure x_t are inputted to the control unit and the overrided control is realized.

By equ(8), the infeedrate is adaptively tuned based on the difference between the estimate and the measure for the constant force grinding, the variance of residuals are also monitored. When the abnormalities take place including wheel blunt and blocking, the residuals increases sharply different from that of the state grinding process.

In light of this rule, the on-line monitoring and adaptive control have been developed. The results are shown as fig 7. The tests confirm that the in process monitoring and control of internal grinding could prevent the thermal damage effectively.

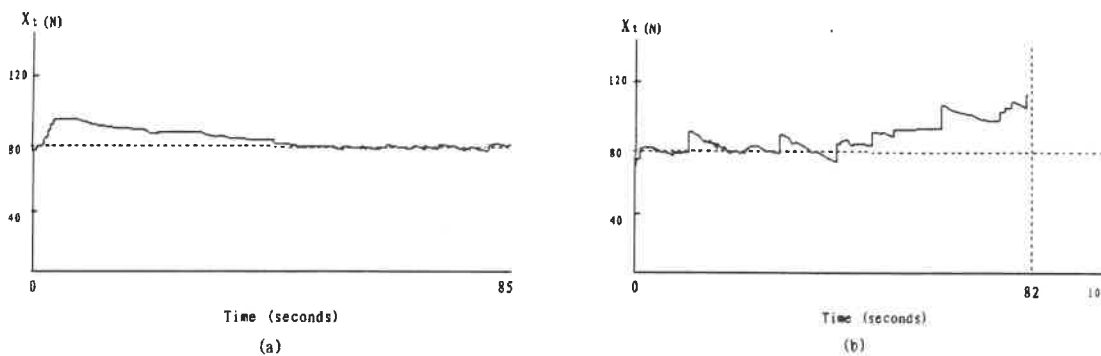


Fig. 7 The difference between the radial force and the tangential force changes in (a) normal process and (b) abnormal process

Conclusion

The internal grinding of deep hole in high strength steel is an advance technology. Special attachment and effective dressing set up have been built up, two major factors relative to the dress are also discussed. Due to the lack of rigidity of the mechanical structure, the deflection of spindle is investigated and relationship between the feedrate and grinding force is highlighted. At the end of this paper in-process monitoring and control of the process are proposed. Based on grinding force, one future step is estimated to control the infeed rate while variation of residuals is used to defect the grinding state including the overcut and surface state of grinding wheel. The grinding thermal damage is prevented effectively.

Conference:

1. Pacherer E. Grinding of Steel with Cubic Boron Nitride (CBN) Annual of CIRP, 34(2), 1985, P 557-563.
2. Zhu Feng. The Inference of Surface State of Resin Bonded CBN Wheel on the Grinding Ratio. Journal of HuaZhong University of Science and Technology. vol 22(2), 1994, P40-44.
3. Toenshoff, H.K., Zingrebe, M. Optimization of Internal Grinding by Microcomputer Based Force Control. Annals of the CIRP, vol. 35(1), 1986, P293-296.
4. Ma Ping, Wang Min. The Quality improvement of Internal Grinding in High Strength Steel, proceedings of international conference on quality and reliability, 1995, vol1, P98-101.

Framework for Assessing Key Variable Dependencies in Loose-Abrasive Grinding and Polishing*

John S. Taylor, David M. Aikens, and Norman J. Brown (retired)
Lawrence Livermore National Laboratory

Summary

This memo describes a framework for identifying all key variables that determine the figuring performance of loose-abrasive lapping and polishing machines. This framework is intended as a tool for prioritizing R&D issues, assessing the completeness of process models and experimental data, and for providing a mechanism to identify any assumptions in analytical models or experimental procedures. Future plans for preparing analytical models or performing experiments can refer to this framework in establishing the context of the work.

Motivation

Precision lapping and polishing processes are likely to be used in manufacturing optics for the National Ignition Facility (NIF). For example, pitch polishing on planetary laps is the key operation that enabled the production of highly flat windows and amplifier slabs for DOE's past and present solid-state ICF lasers: Nova, Omega, Shiva, etc. There are several proposals for developing new high-speed polishing techniques and machinery for reducing the cost of NIF optics. Most of these proposals require significant technical developments before they can lead to cost-effective production processes. It is critical to assess the technical issues associated with these developments.

This paper focuses on the variables that influence the ability of a lapping or polishing operation to meet the figure specification for a workpiece. As an example, a goal for a final polishing operation on a transmissive flat optic might be to attain a final transmitted wavefront of about $\lambda/6$ PV, where λ is the inspection wavelength of 632 nm. For a typical optical glass, this corresponds to a surface flatness requirement of approximately $\lambda/3$ PV, which is about 210 nm. The criterion for determining if a variable has a significant influence on the polishing process is based on whether or not variations in its value lead to significant changes in workpiece flatness.

Approach

Our approach is to employ a well-known tribological model (Preston's Equation) that predicts the local material removal rate in loose-abrasive lapping. We consider how physical variables influence each of the terms in this equation and how part shape is determined by the integration of the local removal rate over the duration of the operation.

Preston's Wear Equation

The model we are using for the material removal rate for loose abrasive lapping operations is the Preston Wear Equation:^{1,2,3,4}

$$\left. \frac{dh(x,t)}{dt} \right|_B = C_B(x,t) P_B(x,t) U_B(x,t) \quad (1)$$

$$\text{removal rate} \propto \text{pressure} \times \text{speed}$$

where h is the height of the surface layer that is worn away, P is the apparent contact pressure between the lap and the workpiece and is equal to the contact load divided by the apparent area of contact, and U is the relative sliding speed between the lap and the workpiece. This relative sliding speed is calculated from the magnitude of the difference in vector velocities of the lap and the part at point B on the workpiece. C is a proportionality constant referred to as Preston's coefficient. Although the equation is written here as a function of position and time, the crux of the statement is simply that *removal rate is proportional to contact pressure and speed*.

* This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract No. W-7405-ENG-48.

If the frictional force exerted by the workpiece along the lap surface is proportional to the contact pressure, then Preston's equation can be interpreted as stating that the rate of material removal $\frac{dh}{dt}$ is proportional to the sliding energy, i.e., *frictional force* $\times$ *sliding velocity*. The constant of proportionality is Preston's coefficient C . For many lapping and polishing operations, C is known, and as long as the conditions remain constant, Eqn. 1 results in a simple and convenient way to estimate removal rates.⁵

Variable dependencies

The influence of all variables on the evolution of surface figure during loose-abrasive processing is determined by the Preston equation. For the cases of relative velocity and pressure, this influence is directly stated by Eqn. 1. For variables such as temperature and material properties, their influence is determined by their effect upon C , P , and U . In this paper, the functional dependence of the local material removal rate will be discussed, with the goal of covering all important variables.

Time

All of the variables involved in lapping can, in general, vary with time. This variation can be repeatable and predictable, such as in periodic variations in the sliding velocity, or they may depend on the evolution of the process, such as pressure variations due to part and lap wear and changes in the properties of the slurry due to aging.

The Framework

Figure 1 shows a diagram that depicts the key functional dependencies for the Preston Equation. Each of the three factors on the right hand side of Preston's Equation has a tree leading to different variables and physical processes that influence it. In order to determine what effect a particular variable has on removal rate, one must first identify where on the chart that variable influences C , P , and U . If that variable is not on the chart, then it is unlikely to influence removal rate. This chart is not intended to be a rigorous description of mathematical dependencies, but to indicate the hierarchy of how physical variables influence the material removal process.

Relative Speed, U

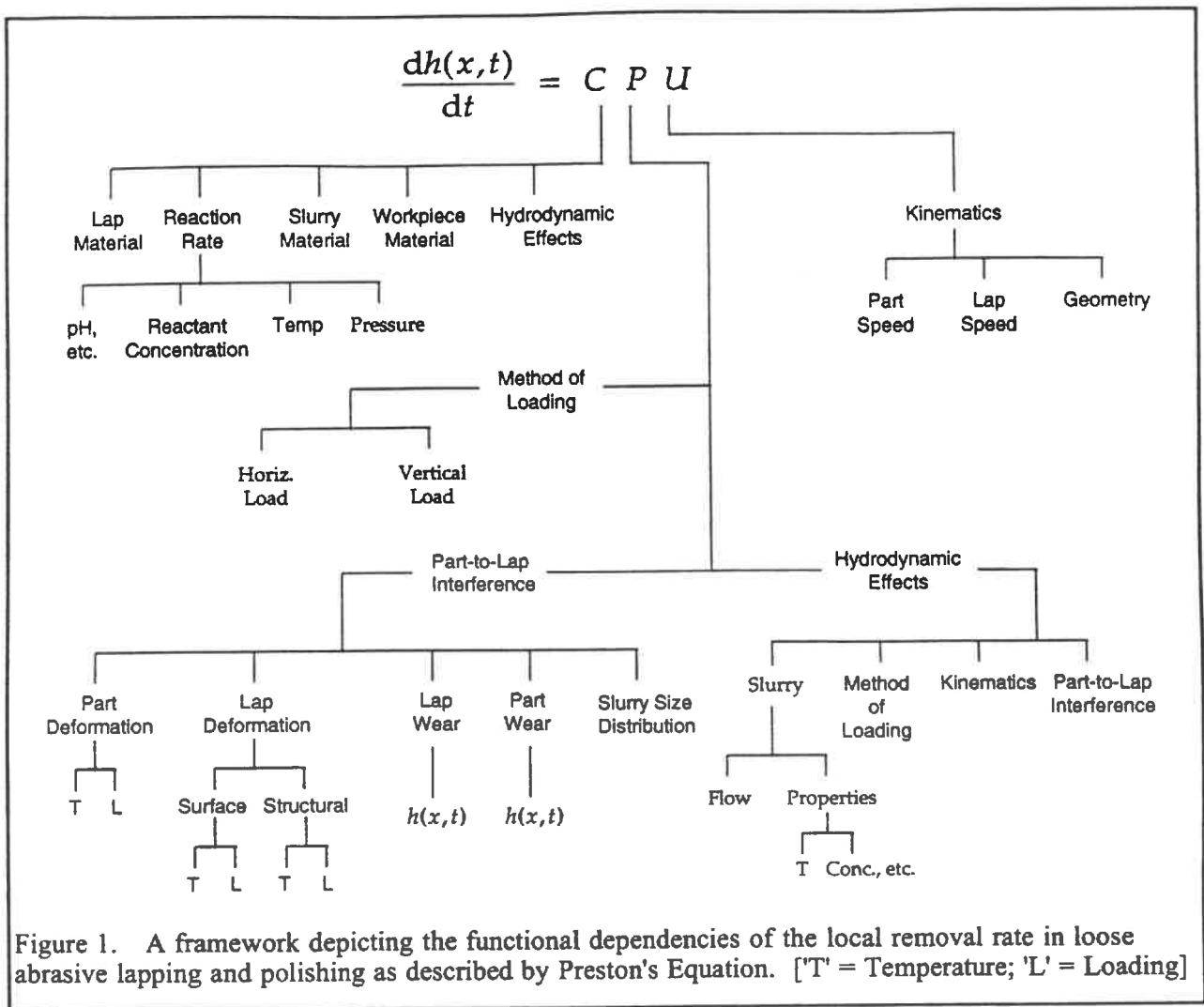
The relative speed between the part and the lap is determined by the magnitude of the vector difference between the lap velocity and the part velocity. In general, the relative speed varies across the part and in time. Geometry is listed here because the influence of velocity on the distribution of velocities depends on parameters, such as where the part is located in the carrier and the style of lap, e.g., overarm or planetary. The influence of kinematic variables in planetary lapping and polishing is discussed elsewhere³, and is not repeated here.

Pressure, P

Pressure is generally not uniform over the surface of the part and will vary as the part moves over the lap. A "non-uniform pressure distribution" refers to different levels of contact force between the part and the lap at different points on the part. Three main categories influence the pressure distribution: 1) part-to-lap interference; 2) method of loading; and, 3) hydrodynamic effects.

A key concept in discussing the pressure distribution on the surface of the part is the "unstressed part shape." Because it is almost always the goal to mount optics in an unstressed state, it is the flatness of the surfaces *in this state* that is important. Thus, if a flat surface is produced on a part in a stressed (and deformed) state, then when it is installed in the system, it will tend to its unstressed shape.

'Part-to-lap interference' refers to all instances where the contact surface between the lap and an *unstressed part* is either not flat or comprises an air gap. When polishing in this mode, the areas with higher pressures will have higher removal rates. The word *interference* is used because it represents the geometric mismatch between the surface of the unstressed part and the lap.⁶



Part-to-lap interference may be caused by several factors: part deformation, lap deformation, lap wear, part wear, and the presence of global variations in the sizes of abrasives on the lap.

Two main causes of part deformation and lap deformation are thermal effects and distortion due to applied loads ('T' and 'L' in the figure). For workpieces, thermal deformation will be determined by the heat transfer boundary (and initial) conditions such as slurry temperature and flowrate, slurry evaporation rates, lap temperatures, material properties, frictional work, etc.

Lap deformation can be divided into localized deformation at the surface and large-scale structural deformation. The structural components are similar to those for the workpiece, but would also include thermal effects that account for any internal cooling (or heating) mechanisms and heat sources such as motors, cooling coils, and bearing friction. Furthermore, the surface of a polishing pad will exhibit localized deformation due to indentation by the "high" spots on part, which might be addressed using a Hertzian analysis.⁷ Also, the properties and geometry of the pad may be temperature dependent.

'Hydrodynamic Effects' refers to a non-uniform bearing-like fluid pressure distribution that develops due to the sliding motion of the part. This can lead to a tilting motion of the workpiece (if it acts as rigid body slider), or cause a net suction (e.g., for concave parts) or lift of the workpiece

with respect to the lap. These influences might be observed as variations in the Preston coefficient. Variables that influence the hydrodynamic effects include the slurry flowrate and its properties, the method of applying traction loads to the workpiece, kinematics, and the part-to-lap interference.

The 'Method of Loading' directly influences the pressure distribution. If the vertical load is applied in a non-uniform distribution, e.g., a small number of discrete loads, then the pressure distribution will tend to a linear (planar) distribution for a rigid-body workpiece, and may lead to a complex distribution if the workpiece deforms. The judicious placement of small weights on the surface of workpieces has long been a key tool used by opticians to correct various flatness errors on workpieces. Workpieces might also be loaded from above by a pneumatically-loaded plate, which can be rotationally-driven. If the motion of this plate or its flatness causes deformation of the workpiece, then the pressure distribution under the workpiece will be affected. In addition, if this plate prevents the workpiece from tilting to follow the lap, such as due to stiction in the gimballing motion of the plate, then a non-uniform pressure distribution will be created. The method of applying the horizontal load on the part will, in general, cause a moment (or force couple) to form with the inertial forces and the frictional forces, contributing to a non-uniform pressure distribution under the part.

Influences on the Preston Coefficient, C

The Preston coefficient accounts for the efficiency in converting frictional energy into material removal. It depends upon the material properties of the lap, the slurry distribution and its properties, and the workpiece properties, including thermal, mechanical, and chemical properties. Because polishing might be viewed as stress-induced chemistry, the properties of the reactants are important. Similarly, any variable that influences the reaction rate can influence the Preston coefficient. Furthermore, Cook⁸ and Brown⁹ point out that a significant amount of the material removed during the polishing of silica glasses is redeposited on the surface, which clearly affects the apparent rate of material removal, and thus C . Variables that might influence the reaction rate include the reactant concentration, the pH of the fluid, the temperature (Arrhenius dependency) and the local pressure.

Coupling of effects

It is clear that there can be substantial coupling among the physical influences upon the material removal rate. Thus, changing a variable such as temperature in lapping and polishing can cause several effects. In comparing different experiments or calculations that investigate the influence of particular variables, we should use the framework of Figure 1 to determine if the link to the local material removal rate is the same. Ensuring a common link to C , P , and U will provide an "apples-to-apples" comparison among different experiments.

¹ Preston, F., "The theory and design of plate glass polishing machines," *J. Soc. Glass Tech.*, vol. 11, pp. 214-257 (1927).

² Brown, N. J., *Optical Fabrication*, UCRL-MISC- 4476 Rev. 1, September 1990.

³ Taylor, J. S., et. al., "High-speed lapping," *1994 Engineering Thrust Area Report*, LLNL, *in press*.

⁴ Cumbo, M. J., *Chemo-Mechanical Interactions in Optical Polishing*, Ph.D. Thesis, University of Rochester (1993).

⁵ For example, for pitch polishing of BK7, C is about $8(10)^{-14}$ cm²/dyne (see ref. 2). Therefore, to calculate the removal rate for a pressure of 25 gm/cm² and a speed of 10 cm/s:

$$\frac{dh(x)}{dt} \equiv 8(10)^{-14} \frac{\text{cm}^2}{\text{dyne}} \times 25 \frac{\text{gm}}{\text{cm}^2} \times 10 \frac{\text{cm}}{\text{s}} \times 981 \frac{\text{dynes}}{\text{gm}} \times 10,000 \frac{\mu\text{m}}{\text{cm}} \times 3,600 \frac{\text{s}}{\text{hr}} = 0.7 \frac{\mu\text{m}}{\text{hr}}$$

⁶ There is an analogy here with the concept of interfering asperities in tribology, see N. P. Suh, *Tribophysics*, Prentice Hall, 1986, chapter 3 "Generation and transmission of force at the interface: the genesis of friction."

⁷ see Suh, N. P. *op cit.*, chapter 4, "Response of materials to surface traction."

⁸ Cook, L. M., "Chemical processes in glass processing," *J. Non-Crys. Solids*, 120 (1990) pp. 152-171.

⁹ Brown, N. J., "Some speculations on the mechanisms of abrasive grinding and polishing," *Precision Engineering*, 9(3), July, 1987, pp. 129-138.

Extended Abstract
N/C Polishing Program
Continued Force Wear and Part Correction
Experiments

P.R. Hannah, R.D. Day, D. J. Hatch (Los Alamos National Lab.)
E.R. McClure (Moore Tool)

INTRODUCTION:

This abstract reports the near completion of the first phase of this program. It is the aim of this program to provide the operator of a N/C diamond turning machine or N/C grinding machine (jig grinder) with the wear characteristics necessary to achieve uniform material removal.

The second phase of this program addresses a different problem, although solving this problem is highly dependent on the results of the first phase. Diamond turned, or any lathe turned surface, exhibits regular tool marks due to the tool passing over the surface being cut. Changes in depth of cut, feed rate and work rpm will change the character of these grooves, but will not eliminate them. Optical surfaces produced by this process exhibit increased scattering as the light wavelength decreases limiting their use; at least for optical purposes, to IR and some visible applications.

Utilizing wear information gathered in the first part of this program we will attempt to reduce these residual tool marks by polishing. The polishing of diamond turned surfaces is not new. Diamond turned metal surfaces, especially in electroless nickel and high phosphorus nickel electroplate have been polished to improve their scatter characteristics. What we believe is unique is the use of a spherical wheel, rotating on axis and being moved over the part in a prescribed manner by numerical control.

Over the past year we have made some major changes in our polishing methods and procedures. We have listed below these changes, as a refresher for the reader as to our previous procedures [1]. These changes will be addressed in the body of the text.

Previously reported	Current procedure
1. •Polyurethane wheel w/ partial <u>toric</u> contour, maj. radius 200 mm, minor radius 18 mm.	• <u>Spherical</u> polyurethane wheel 19 mm in Diameter. Speed and rotation as before (1000rpm-CW).
2. •Periphery of the wheel used to polish.	•Rotational <u>axis of sphere</u> used to polish (FIG.-1).
3. •Polishing slurry <u>dripped</u> at wheel/work interface.	•Polishing slurry applied by <u>brush</u> at, and ahead of wheel/ work interface (FIG.-2)
4. •Part spindle speed <u>100rpm</u> .	•Part spindle speed <u>30rpm</u> .
5. •Polishing compound Rhodite 906	•Polishing compound Rhodite 904

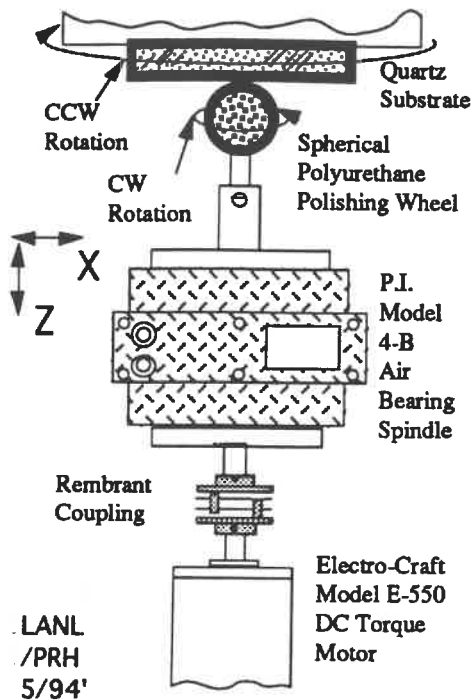


Figure 1 - Spherical Polishing Wheel Arrangement

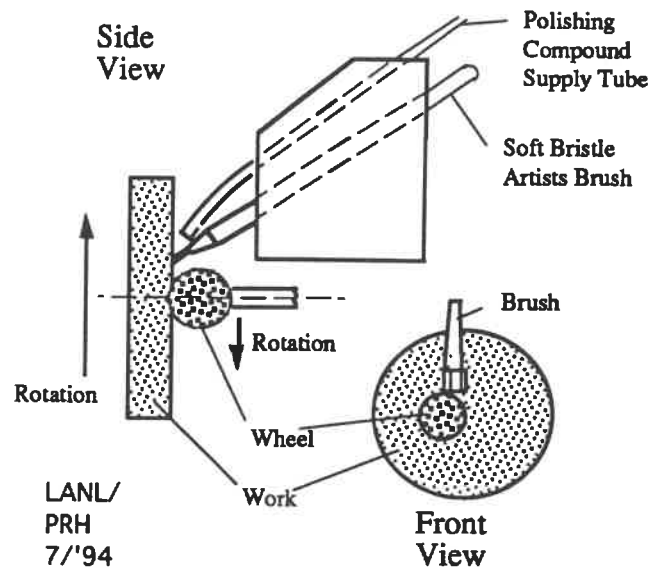


Figure 2- Polishing Compound Distribution

PROGRESS:

In our poster paper which was presented at the 1993 ASPE meeting [1] we reported the following conclusion. "A model has been developed, using Preston's equation to predict the shape of a work piece after polishing on a NC/DTM. The model predicts the workpiece's general shape but does not accurately predict the material removal. Additional work is needed to resolve this discrepancy."

During the course of the year it became obvious that there were indeed areas in our experimental procedure that needed to be changed. Listed above are the major changes that we have incorporated into the process. These changes have improved the process to the point where we obtain nearly repeatable results (FIG.-3). There are still some elusive wear factors that need to be added to the mathematical model as the theoretical material removal (wear) is about 30% above the measured removal, a significant improvement over last years reported factor of 2.

We have recently corrected a quartz surface which had been used for our wear tests (FIG.-4). The deep area at the edge is produced by the polishing wheel dwelling at this point. The polyurethane polishing medium (Rhodes "Greyrock") at 5 Newtons exhibits some viscoelastic properties. At this pressure, time must be taken to allow the wheel to stabilize, thus producing a uniform polishing pressure, hence the dwell zone.

We also need to point out that in our current tests the wheel; which travels from the edge to the center, is stopped 5mm from the center.

FIG.-5 shows the part prior to polishing. Ignoring the edge and center anomalies the part deviates from the flat about $\lambda/5$ ($\lambda = 632.8 \text{ nm}$). After

polishing the part deviates from the flat about $\lambda/15$. This factor of three improvement on fractional wavelength surfaces gives us confidence as we move to diamond turned metal surfaces in the second phase of this program. The RMS surface finish on the part before polishing was measured to be 1.68 nm, after polishing it was 2.09 nm. This slight degradation of surface finish was expected because of the aggressive nature of the polishing compound and wheel.

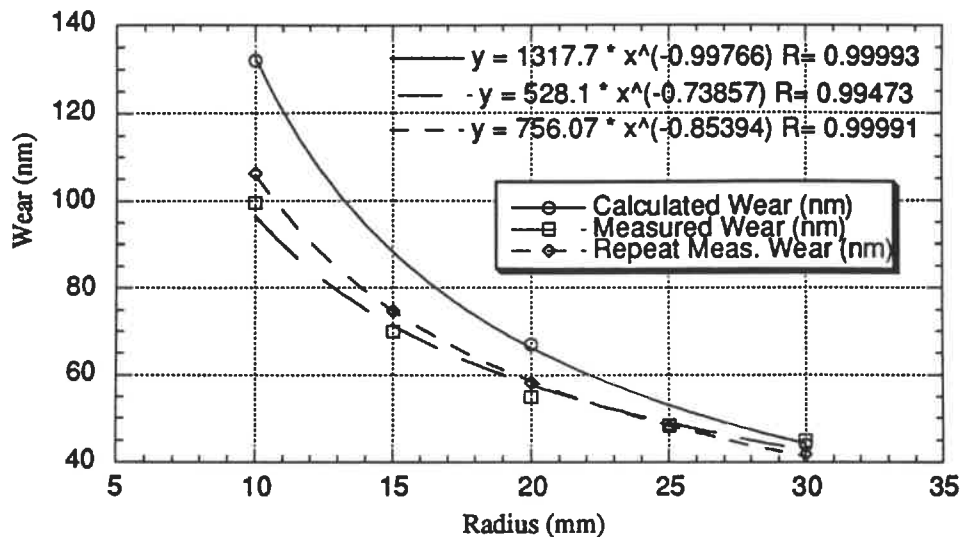


Figure 3 - Measured and calculated values of wear on the quartz surface

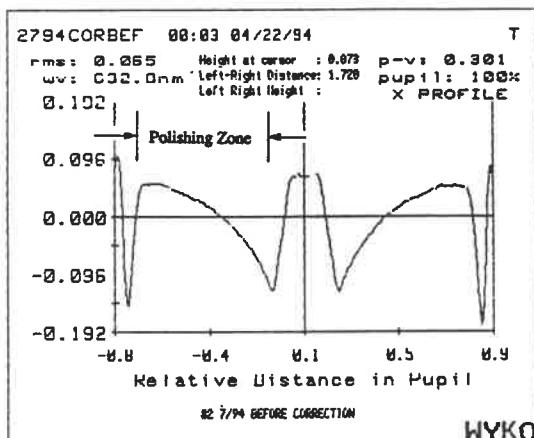


Figure 4 - Section of part shape before polishing

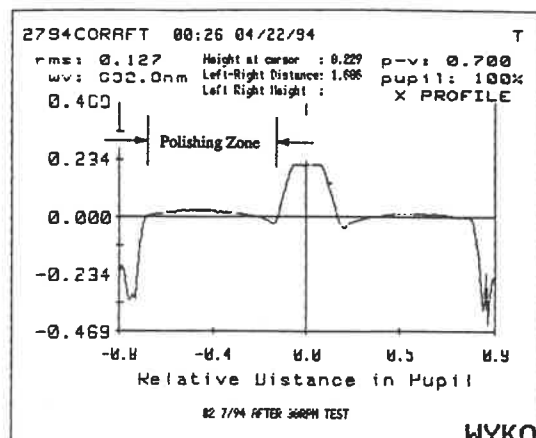


Figure 5 - Section of part shape after polishing

DISCUSSION OF IMPROVEMENTS:

1.&2. The large contact area of the toric wheel was seen as a problem especially as related to the center of the part. A spherical wheel, rotated on its axis seemed to be a better solution, as its contact area, at the pressures required, was only 2mm. The spherical wheel also is better suited to the polishing of complex internal forms, and with a simple change in the wear equation, can be tilted to accommodate different grinding head configurations.

3.&5. In looking closely at the compound distribution problem we could see that the small bore of the hose made its application very critical. A slight lateral miss adjustment of the hose would cause the compound to fail to find the interface between the wheel and work. The meniscus that was formed between the compound drop and the work was also affected by the change in radius as the hose moved across the work. FIG.-3 illustrates the solution to this problem.

The brush technique is nothing new, as one of the authors saw it being used in the 1960's at Perkin-Elmer. The diagram should be self explanatory. We would point out that having the trailing edge over the point of contact insured ample, fresh compound was being supplied to the polishing interface at all times. In actuality this solution made a major improvement in repeatability from part to part.

The change in compound from Rhodite 906 to 904 was due to the optical shop's supply of 906 being exhausted. A call to Rhodes confirmed that the only difference between the two compounds was one of settling time. We have observed that the 904 seems to provide more uniform wetting. Both compounds average particle size are $2\mu\text{ m}$.

4. While observing the compound distribution problem we saw that centrifugal force acting on the compound seemed to thin it, and sling it off at the edge of the part. The compound that reached the center of the part seemed to build up to a point where it too found a non-uniform radial path to the edge. A simple calculation showed us that a reduction in rpm from 100 to 30 gives a factor of ten reduction in the centrifugal force at the edge. This reduction has also made a significant difference in test repeatability.

CONCLUSIONS:

We have developed an understanding of the N/C polishing process, performed on a Diamond Turning Machine, to the point where it can be repeated to the repeatability of our interferometer (6 nm). We are now able to predict the amount of material removal to within 30% of the measured values. By using the model we have been able to correct a part from $\lambda/5$ to $\lambda/15$ (where $\lambda = 623.8\text{ nm}$).

ACKNOWLEDGEMENTS:

Funding for the LANL portion of this work was provided through the Technology Transfer Initiative of the Department of Energy. We appreciate Rodney Schmell's provision of the polishing blanks. Two of the authors, RDD and PRH, appreciate the opportunity to study "the fixed laws of nature" which God established and apply them towards the betterment of mankind.

REFERENCES:

[1] P. Hannah, et. al., Proceedings ASPE 1993 Annual meeting, pp. 262-267.

A COMPUTER-CONTROLLED POLISHING MACHINE FOR STRONGLY ASPHERIC OPTICS

Qiang Kong, Robert Zimmermann, and Martin High

Applied Physics Specialties Ltd.
17 Prince Andrew Place, Don Mills, Ontario, Canada M3C 2H2

Introduction

As the demand for compact light weight optical systems increases, optical designers are including refracting glass aspheres in their designs.

Most optical systems can be successfully corrected using spherical surfaces. However, as specifications become more demanding the number of elements rises, weight increases and transmission falls.

The introduction of aspheric surface reduces weight, increases transmission and reduces the number of mechanical parts.

The higher cost of aspheric surfaces can be reduced with the use of automation and other innovative manufacturing processes and as costs fall, new applications become feasible.

APS has developed a process to fabricate highly aspherical lenses using diamond grinding followed by computer controlled polishing.

Equipment Description

A computer-controlled polisher (CCP) has been constructed using off-the-shelf precision positioning system and other commercially available mechanical components. The polishing head is mounted on a pressure-sensitive polishing arm that is mounted on a linear stage. A subaperture tool is moved across the optical surface radially, while the part is mounted on a spindle for circularly symmetric element. First the surface metrology data was obtained by Form Talysurf Series contact profilometer. The relative dwell time of tool over annular position was then calculated with a mathematical algorithm based on its deviation from nominal, the pressure of tool applied on part, and other operation parameters. By varying the amount of time the machine works any region, a controlled amount of materials may be removed. The best tool configuration for any figuring operation is determined by use of computer modeling, while the proper operating parameters are obtained from experimentation.

Polishing Process and Results

The field corrector is a critical part of head-up displays, which provides flight information in military as well as commercial aircraft. However, its large aspheric departure from the best fit sphere (1-2mm) makes it impractical to be fabricated using classical techniques in medium-volume production. That makes it an ideal example for the computer controlled polishing.

A demonstration was conducted to show the ability of the CCP to fabricate such a strong asphere. A 8 inches diameter, 0.7 inch thick SK-2 lens was used as test piece. It has the following aspherical coefficients: $RV = 259.46235$, $K = 0$, $A = -4.69498E-8$, $B = -9.98370E-14$, $C = -4.15788E-17$, $D = 1.68939E-21$. The deviation from the best fit sphere is shown in Fig.1 with the maximal deviation of 1.35mm.

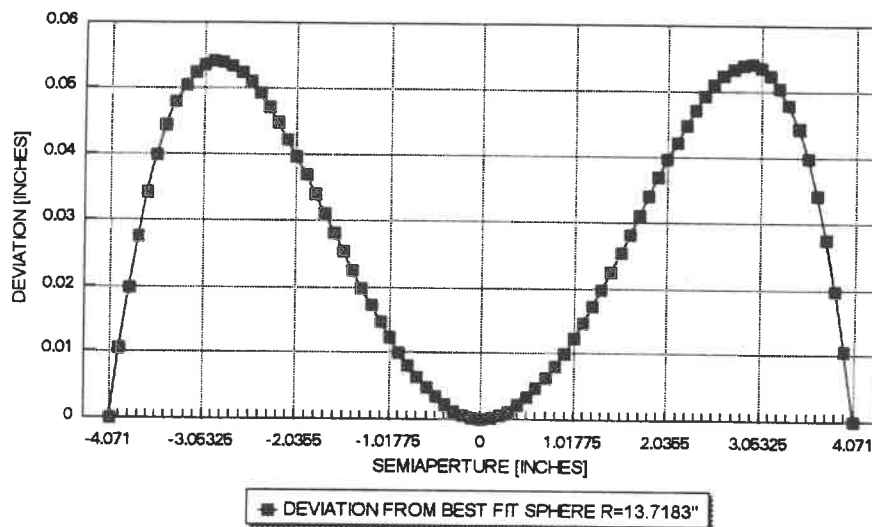


Figure 1. Figure deviation from the best fit sphere of an 8 inches field corrector.

The part is pre-polished using conventional techniques or on the CCP. The pre-polishing time is equal to what it takes for polishing sphere using conventional techniques. The figure is assessed using Form Talysurf Series contact profilometer. Fig. 2 shows traces from centre to edge of the lens at three polishing stages. The top graph shows a peak-to-valley of more than 2 micron deviation from its nominal, a typical result from the pre-polished process.

This profile data is analyzed and used to generate a control code for regulation of the CCP. The machine varies the velocity of the polishing assembly along the polishing path, causing the desired varying material removal.

The total time per figuring cycle was selected so that it is short enough to avoid profile errors due to tool wear during each polishing cycle, and long enough to allow a large ratio between the

dwelt time at the edge and the centre. It may also be found necessary to introduce empirical correction to the calculation of dwelt time.

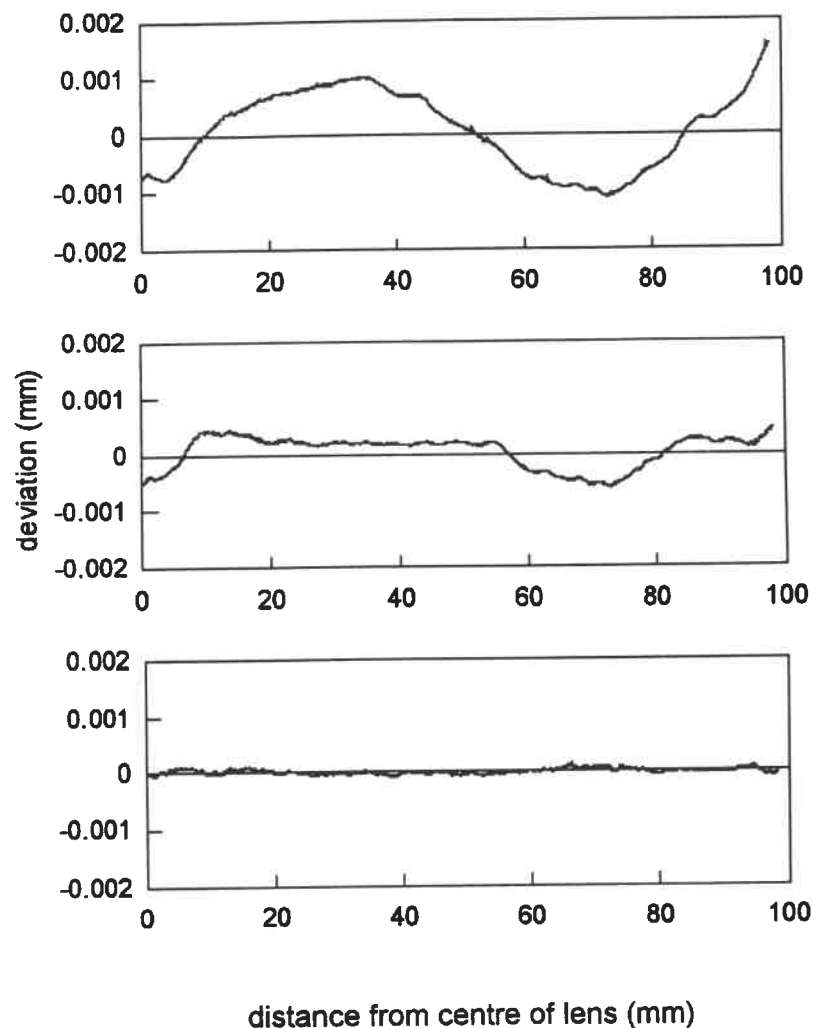


Figure 2. Traces of the central part of an aspherical lens; top, after pre-polishing using traditional technique; middle, after one figuring cycle on the CCP; bottom, final profile after several cycles of CCP polishing.

After the desired amount of polishing, the workpiece surface is tested again. The peak-to-valley value drops to 1 micron (middle). By using the iterative Talysurf testing and figuring, the workpiece surface can converge to the desired figure. Following several additional figuring cycles, the final figure of 0.3 micron was achieved (bottom). This was well within the design tolerance of 1 micron. The total figuring time is such that the process is commercially viable.

Fabrication of over 20 pieces of these field corrector and some other aspheric lenses have proven the consistency of this process.

Conclusions

A recent fabrication effort has demonstrated the ability of this process to manufacture field correctors with both accurate figure (< 1 micron peak-to-valley) and smooth surface in minimal time and thus a more affordable price. The CCP has been used also to fabricate a number of other difficult aspheric lenses. The technology points towards further reduction in figuring time.

The authors wish to acknowledge the useful discussion and support of colleagues at APS.

Determinism in Dice Throwing and the Transition to Chaos

Layton C. Hale
Massachusetts Institute of Technology
Cambridge, MA

Introduction

Determinism is both a principle and a philosophy, which applies to many systems and processes such as automated manufacturing processes and machine tools in particular. As a principle, determinism rests on the ability of classical physics to explain the behavior of machines and processes. As a philosophy, determinism instills a belief that all aspects of a machine or process can be understood and ultimately controlled as desired. Beyond that, determinism applies to precision machines that have yet to be conceived, and so too it is a principle and a philosophy of design. For purely educational motives, we will test the practical limits of determinism by exploring a process that conventional wisdom holds as probabilistic. Specifically, we will determine the level of precision required to control the trajectory of a die to a predictable state of rest.

James Bryan described dice throwing as a completely deterministic process, which appears random even though the laws of motion are not probabilistic¹. As a matter of principle, determinism applies. As a practical matter, the effects of errors compound with each bounce so that seemingly insignificant errors could result in unpredictable behavior. In addition, there are unstable states of rest (on an edge or a corner) where an infinitesimal influence could topple the die onto one of several faces. Such states are unpredictable and corresponding trajectories must be avoided. Is this system an example of chaos, and is chaos an exception to determinism? After all, the

mathematics of chaos was generally unknown until 1976, well after Bryan wrote his paper in 1971.²

To answer these questions, we will develop a computer model to simulate the die's trajectory and use it to develop an average sensitivity model that relates variations in the die's orientation after one bounce to error sources such as the initial state of the die and the parameters that affect the die-table interaction. As these changes will affect the next bounce, the sensitivity will increase exponentially with the number of bounces, and at some point the trajectory will become uncontrollable from initial conditions alone. This is characteristic of a chaotic system like fluid flow that becomes turbulent at sufficiently high Reynolds number. Using this sensitivity model, we will estimate the acceptable errors for a case having a reasonable number of bounces. Then we will demonstrate through simulations that the process is largely deterministic provided that these tolerances are achievable in practice.

Model Development

The system of coordinates shown in Figure 1 consists of an inertial x-y-z system fixed to the table and a moving 1-2-3 system fixed to the die. The 1-2-3 system aligns respectively to the 1-2-3 faces, and the 4-5-6 faces are arbitrarily chosen to be opposite of the 1-2-3 faces, respectively. The motion of the die is represented by a twelve component state vector consisting of the position of the mass centroid

¹ J. Bryan "The Power of Deterministic Thinking in Machine Tool Accuracy" UCRL-91531, 1984.

²Over one-hundred books have been written on the topic of chaos. See for example, Ian Stewart "Does God Play Dice?: the Mathematics of Chaos" 1989. A common view of chaos is that some nonlinear dynamic systems exhibit such complex behavior as to be unpredictable without perfect knowledge of the initial conditions and of the coefficients in the differential equation. This view does not conflict with the principle of determinism.

$\mathbf{r}$, the angular orientation of the 1-2-3 system, the velocity $\mathbf{v}$ and the angular velocity $\boldsymbol{\omega}$. The angular orientation is uniquely defined by a 3 by 3 rotation matrix that can be developed from sequential rotations about three axes (e.g., θ_x then θ_y then θ_z) or from a single rotation about an angle vector $\boldsymbol{\theta}$. These representations are different from one another and are different from simultaneous rotations about three axes that result from the time integration of angular velocity. The rotation matrix is indispensable for transforming point locations, velocities and angular velocities from one coordinate system to the other.

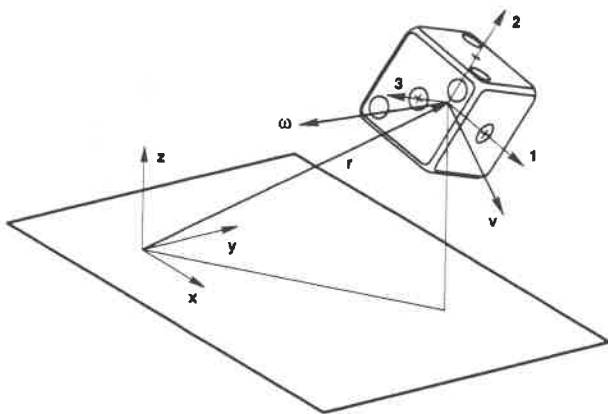


Figure 1: The system of coordinates consists of an inertial x-y-z system fixed to the table and a moving 1-2-3 system fixed to the die.

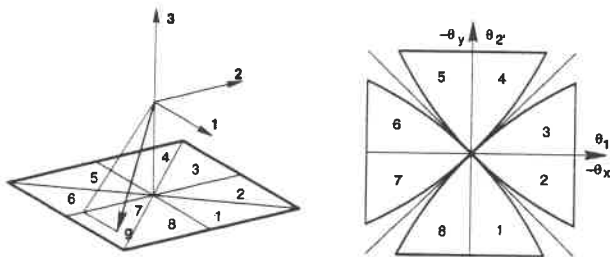


Figure 2: The orientation of the gravity vector through any face has an eight-fold symmetry represented in 3D (left) or projected into angular coordinates (right).

It is important in a large and complicated problem to reduce the number of parameters to test. For example, the symmetry of the table is such that the initial conditions for x , y and θ_z will not affect the simulation. In addition, any possible trajectory can be started from an initial condition where v_z is zero. If we also set the remaining *nominal* velocities to zero, then as shown in Figure 2, the range of angular initial conditions takes on an eight-fold

symmetry for any face. While the results will obviously not be symmetric (i.e., face numbers), the *ideal* trajectories will be symmetric. Thus without any real loss in understanding the system, we will restrict the angular range to octant 7 where $0 \leq \theta_x \leq 45^\circ$ and $0 \leq \theta_y \leq \arctan(\sin(45^\circ))$.

The trajectory of the die through space is calculated using a fourth-order Runge-Kutta algorithm to integrate the system of nonlinear state equations given in (1). Rows 1 and 2 are the kinematic equations. Row 2 contains the Jacobian matrix $\mathbf{J}$, which relates the die's angular velocity to the rates of change of sequential angles that describe the die's orientation. With a little effort, the Jacobian can be derived from Equation 2, where each side describes the die's changing orientation.³ Rows 3 and 4 are the kinetic equations and include the acceleration of gravity $-\mathbf{g}$ and quadratic terms for aerodynamic drag.⁴

$$\frac{d}{dt} \begin{bmatrix} \mathbf{r} \\ \boldsymbol{\theta} \\ \mathbf{v} \\ \boldsymbol{\omega} \end{bmatrix} = \begin{bmatrix} \mathbf{v} \\ \mathbf{J}(\boldsymbol{\theta})^{-1} \cdot \boldsymbol{\omega} \\ -\mathbf{g} - \frac{b}{m} \|\mathbf{v}\| \mathbf{v} \\ -\frac{c}{I} \|\boldsymbol{\omega}\| \boldsymbol{\omega} \end{bmatrix} \quad (1)$$

$$\frac{d}{dt} [\mathbf{R}(\boldsymbol{\theta})] = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix} \cdot \mathbf{R}(\boldsymbol{\theta}) \quad (2)$$

A nonlinear solver based on Newton iteration terminates the integration when the die makes contact with the table. At this instant an impact model calculates new velocity states for the die, and either the integration will start over, or the simulation will terminate if there is

³To avoid orientations where the Jacobian matrix becomes singular and to speed the numerical integration, an intermediate coordinate system, aligned after each bounce to the spin axis of the die, was used in the simulation to track the orientation.

⁴The fourth row is simplified to take advantage of a cube having equal principal moments of inertia; however, the simulation is more general and accepts three separate principal moments of inertia.

insufficient energy to change to another face. The table, however, is assumed to be immobile since this effect can be lumped into a coefficient of restitution.

To derive the impact model, we start with the kinematic relationship given by Equation 3 between the velocity at the contact point $\mathbf{v}_a$ and the velocity states of the die. The transformation matrix $\mathbf{A}$ defined in (3) also appears in the kinetic relationship given by Equation 5 between the impulse δ_a and the change in velocity states. Combining these equations in (6) results in an invertible relationship between $\mathbf{v}_a$ and δ_a , where $\mathbf{C}$ is a diagonal matrix involving reciprocals of the mass m and the moment of inertia I .

$$\mathbf{v}_a = \mathbf{A} \cdot \begin{bmatrix} \mathbf{v} \\ \omega \end{bmatrix} \quad (3)$$

$$\mathbf{A} = \begin{bmatrix} 1 & 0 & 0 & 0 & r_{a_z} & -r_{a_y} \\ 0 & 1 & 0 & -r_{a_z} & 0 & r_{a_x} \\ 0 & 0 & 1 & r_{a_y} & -r_{a_x} & 0 \end{bmatrix} \quad (4)$$

$$\begin{bmatrix} m \Delta \mathbf{v} \\ I \Delta \omega \end{bmatrix} = \mathbf{A}^T \cdot \delta_a \quad (5)$$

$$\Delta \mathbf{v}_a = \mathbf{A} \cdot \mathbf{C} \cdot \mathbf{A}^T \cdot \delta_a \quad (6)$$

The first step in the impact algorithm is to calculate the impulse required to change the contact velocity to zero. Slip will occur in the plane of the table when the angle between the impulse vector and the table normal is greater than the friction angle $\tan(\mu)$, where μ is the coefficient of friction. The algorithm handles

slip by directing the impulse vector along the friction angle and by changing its magnitude so that the normal contact velocity remains zero. The elasticity of the die and the table is modeled using the coefficient of restitution e , which is fairly crude but intuitive. The effect is to enlarge the impulse by a factor $(1 + e)$, where $e = 0$ for a perfectly plastic material and $e = 1$ for a perfectly elastic material. The new velocity states are then calculated using Equation 5.

Sensitivity Analysis

We will choose a set of *reasonable* parameters to conduct the sensitivity analysis realizing that the results are highly dependent on these parameters. The die will have a vertical velocity as if it were dropped from a height of 0.2 m. The angular orientation will range over one octant to obtain an average sensitivity. The coefficient of friction and the coefficient of restitution for a semihard surface would be on the order of 0.2 and 0.6, respectively.

After the first bounce, the average velocity vector is $[-0.3633, 0.1781, 0.3776]$ m/s, and the average angular velocity vector is $[-66.448, -122.82, 0]$ r/s. The variation in the angular orientation at the next bounce is given approximately by Equation 7, where $2v_z/a$ is the approximate time to the next bounce. Equation 8 shows the sensitivity of the velocity states to small variations in the state vector, the coefficient of friction and the coefficient of restitution. The angular orientation and the coefficient of friction are very significant terms. By substituting into (7) only those terms in (8)

$$\begin{bmatrix} d\theta_x \\ d\theta_y \end{bmatrix} \cong \frac{2dv_z}{a} \begin{bmatrix} \omega_x \\ \omega_y \end{bmatrix} + \frac{2v_z}{a} \begin{bmatrix} d\omega_x \\ d\omega_y \end{bmatrix} \quad (7)$$

$$\begin{bmatrix} dv_x \\ dv_y \\ dv_z \\ d\omega_x \\ d\omega_y \\ d\omega_z \end{bmatrix} = \begin{bmatrix} 0.9096 & 0.0577 & 0.1964 & 0.0010 & 0.0018 & 0.0004 & 0.4420 & 0.9661 & 2.0564 & 0.2365 \\ 0.0577 & 0.8409 & 0.1062 & 0.0021 & 0.0010 & 0.0007 & 0.9186 & 0.5038 & 1.1115 & 0.1279 \\ 0.0835 & 0.0448 & 0.3333 & 0.0025 & 0.0043 & 0.0000 & 1.2709 & 2.4147 & 1.9466 & 1.4762 \\ 17.064 & 59.949 & 43.262 & 0.4594 & 0.1752 & 0.1964 & 249.70 & 75.352 & 203.43 & 52.232 \\ 48.675 & 17.064 & 70.426 & 0.1752 & 0.5016 & 0.1062 & 76.129 & 347.27 & 392.60 & 84.955 \\ 10.038 & 18.227 & 0.0000 & 0.1964 & 0.1062 & 0.9067 & 0.0000 & 0.0000 & 0.0000 & 0.0000 \end{bmatrix} \cdot \begin{bmatrix} dv_x & dv_y & dv_z & d\omega_x & d\omega_y & d\omega_z & d\theta_x & d\theta_y & d\mu & de \end{bmatrix}^T \quad (8)$$

attributed to the orientation, we obtain a prediction for the sensitivity after n bounces given by Equation 9. This equation can be used in reverse to estimate the average angular precision required for deterministic behavior. Equation 8 provides a way to scale this estimate to obtain estimates for the other error sources.

$$\begin{bmatrix} d\theta_x \\ d\theta_y \end{bmatrix}_n \cong e^{2(n-1)} \begin{bmatrix} 36.4 & 38.5 \\ 37.7 & 87.2 \end{bmatrix}^n \cdot \begin{bmatrix} d\theta_x \\ d\theta_y \end{bmatrix}_0 \quad (9)$$

Simulation Results

The simulations shown in Figures 3, 4 and 5 have the same nominal parameters as the sensitivity analysis; however, the angular range is much smaller and centered about $\theta_x = 30^\circ$ and $\theta_y = 20^\circ$. Each grid point in a figure represents one simulation, and its result is indicated on the vertical scale. The results in Figure 3 appear random but a closer look in Figure 4 reveals deterministic behavior. This begins to break down in Figure 5 with the addition of uniform random errors of magnitude $dz = 0.00030$ m, $d\mu = 0.000020$ and $de = 0.000050$.

Conclusion

The sensitivity analysis and the results of the simulations indicate that it is impractical to control the process of dice throwing by initial conditions alone. We have plenty of precision engineering power to control the initial conditions well enough, but the die-table interaction typically has two to three orders of magnitude greater variability than required for a repeatable process. This is not, however, a defeat for the principle of determinism. We now know what is required to make the process deterministic. We could use a passive approach and seek to find the right combination of materials with sufficient control of their properties, surface textures, contamination levels, dimensions and so forth to achieve the tolerances required for μ and e . Alternatively, we could apply an active approach and use closed-loop feedback to control the process by using, for example, a microchip radar to track the die and a smart material on the table to affect the coefficient of restitution.

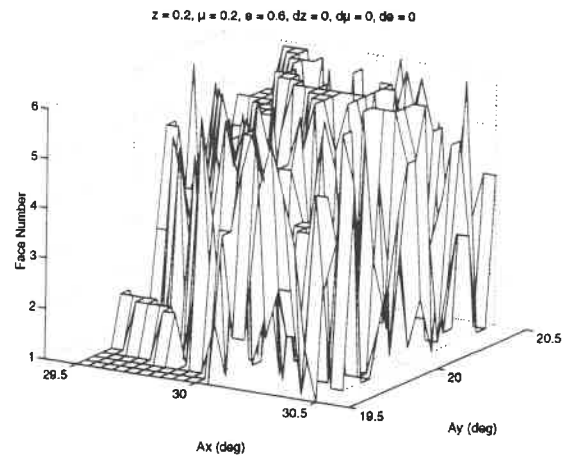


Figure 3: The simulation over a fairly small angular range of $30^\circ \pm 0.5^\circ$ by $20^\circ \pm 0.5^\circ$ shows apparently random behavior over most of the range.

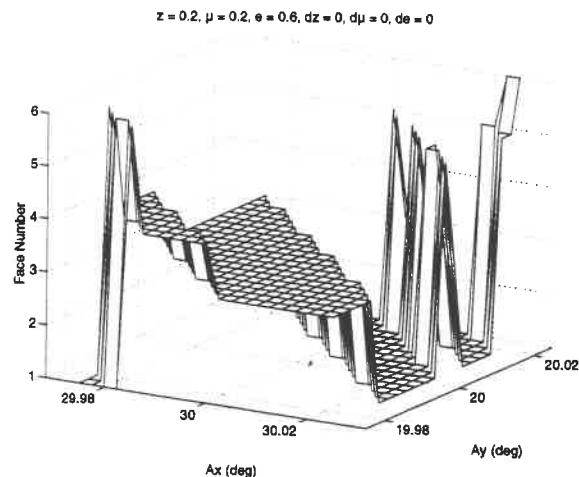


Figure 4: This simulation over a very small angular range of $30^\circ \pm 0.025^\circ$ by $20^\circ \pm 0.025^\circ$ shows the angular precision required for deterministic behavior.

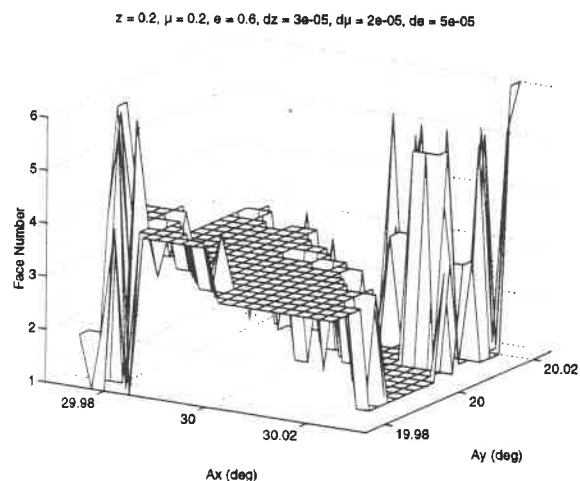


Figure 5: Uniform random errors of order 10^{-5} added to the simulation cause non repeatable behavior near regions of transition. By 10^{-4} , all repeatability is lost.

Analytical Determination of the Temperature Distribution Within a Hollow Hemisphere During a Machining Process

J.C. Shipley
Los Alamos National Laboratory
Los Alamos, NM

G.P. Mulholland
New Mexico State University
Las Cruces, NM

Los Alamos National Laboratory is actively involved in research to determine effective procedures for the accurate machining of hollow hemispheres without cutting fluid. The purpose of this effort is to reduce the amount of mixed waste (cutting fluid and material) which is produced by the machining process. However, this dry-cutting process causes the local temperature of the material to rise and the part to expand. Since this process is one of high accuracy, even modest deformations may be unacceptable.

The goal of this analysis is to analytically determine the temperature distribution of a hemisphere during a typical dry-machining process. The three-dimensional transient energy equation is used as a starting point. At the boundary, the tool-work piece interaction is modeled as an infinite number of discrete heat inputs as functions of position and time on the exterior surface.

Results from this analysis will be verified experimentally and reported in a future paper. The temperature profiles generated from the work will then be used as input for the thermal stress model of the hemispherical shell.

INTRODUCTION

The machining process on any material produces a temperature distribution within the work piece, cutting tool, and the removed material. Usually, most of this energy is dissipated by coolants during the machining operation. For this work, we are concerned with determining the temperature distribution in a hollow hemisphere while it is being machined. In addition, the machining process must be highly accurate and can not use any coolant.

The dry-machining (no coolant) constraint is due to environmental considerations. Since many of the materials being machined are considered hazardous, it is highly desirable to reduce or eliminate the amount of mixed waste (material and coolant) produced during this operation. However desirable this reduction of mixed waste may be, it does introduce problems. The dry-cutting process causes the local material temperatures (cutting tool-material interface) to rise and the part to expand. This in turn creates a stress distribution in the area. Since this is a highly accurate machining operation, even modest deformations may be unacceptable.

The purpose of this work is to develop an analytical expression for the temperature distribution in a hollow hemisphere during a typical dry-machining process. The three-dimensional transient energy equation is used as starting point. Current plans are to initiate an experimental effort to validate this thermal model. Once the thermal model is verified, it will be used as input to a thermal stress model for the hemispherical shell.

THERMAL MODEL

To obtain the thermal model for a hemispherical shell during a typical dry-machining process we start with the three-dimensional transient energy equation in spherical coordinates. At the boundary surface, we consider the tool-workpiece interaction to be an infinite number of discrete heat inputs as functions of position and time. Mathematically, the physical system is modeled in the following manner:

Energy Equation:

$$\frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} \left[\sin \theta \frac{\partial T}{\partial \theta} \right] + \frac{1}{r^2 \sin^2 \theta} \frac{\partial^2 T}{\partial \phi^2} = \frac{1}{\alpha} \frac{\partial T}{\partial \tau} \quad (1)$$

$$a \leq r \leq b \quad 0 \leq \phi \leq 2\pi \quad 0 \leq \theta \leq \frac{\pi}{2} \quad \tau > 0$$

Boundary Conditions:

$$\begin{aligned} \text{a) } T(a, \theta, \phi, \tau) &= 0 \\ \text{b) } \frac{\partial T(b, \theta, \phi, \tau)}{\partial r} &= \frac{Q_0}{k} \delta(\phi - \phi_0(\tau)) \delta(\theta - \theta_0(\tau)) \\ \text{c) } T(r, 0, \phi, \tau) &\text{ bounded} \\ \text{d) } \frac{\partial T(r, \frac{\pi}{2}, \phi, \tau)}{\partial \theta} &= 0 \\ \text{e) } T(r, \theta, \phi, \tau) &= T(r, \theta, \phi + 2\pi, \tau) \\ \text{f) } \frac{\partial T(r, \theta, \phi, \tau)}{\partial \phi} &= \frac{\partial T(r, \theta, \phi + 2\pi, \tau)}{\partial \phi} \end{aligned} \quad (2)$$

Initial Condition:

$$T(r, \theta, \phi, \tau) = 0 \quad (3)$$

In equations (1), (2), and (3) $T(r, \theta, \phi, \tau)$ represents the temperature rise above ambient and

$\alpha \sim$ thermal diffusivity, cm^2/sec

$k \sim$ thermal conductivity, W/cm K

$Q_0 \sim$ heat input to workpiece at tool - workpiece interface, W/cm^2

$\phi_0(\tau), \theta_0(\tau) \sim$ represent the tool locations on exterior surface at time τ

Equation (2b) represents the heat input to the work piece when the cutting tool is located at $r = b$, $\theta = \theta_0(\tau)$, and $\phi = \phi_0(\tau)$. We assume it has magnitude Q_0 (W/cm^2).

The solution for equation (1) subject to the boundary and initial conditions given by equations (2) and (3) can be obtained by the integral transform techniques. The results after much work is found to be

$$\begin{aligned}
T(r, \theta, \phi, \tau) = & \sum_{n=0,2,4,\dots}^{\infty} \frac{2n+1}{2\pi} \left\{ \sum_{l=1}^{\infty} \alpha \hat{H}_{0\cos}(\lambda_{nl}, \gamma_n, m, \tau) * e^{-\alpha \lambda_{nl}^2 \tau} \frac{R_{nl}(r)}{[R_{nl}(r), R_{nl}(r)]} \right\} P_n(\cos \theta) \\
& + \sum_n \sum_m \frac{2n+1(n-m)!}{\pi(n+m)!} \left\{ \sum_{l=1}^{\infty} \alpha \hat{H}_{\cos}(\lambda_{nl}, \gamma_n, m, \tau) * e^{-\alpha \lambda_{nl}^2 \tau} \frac{R_{nl}(r)}{[R_{nl}(r), R_{nl}(r)]} \right\} P_n^m(\cos \theta) \cos(m\phi) \\
& + \sum_n \sum_m \frac{2n+1(n-m)!}{\pi(n+m)!} \left\{ \sum_{l=1}^{\infty} \alpha \hat{H}_{\sin}(\lambda_{nl}, \gamma_n, m, \tau) * e^{-\alpha \lambda_{nl}^2 \tau} \frac{R_{nl}(r)}{[R_{nl}(r), R_{nl}(r)]} \right\} P_n^m(\cos \theta) \sin(m\phi) \\
& - \frac{Q_0}{k} (a-r) P_n(\cos \theta_0(\tau)) \\
& - \frac{Q_0}{k} (a-r) \cos m\phi_0(\tau) P_n^m(\cos \theta_0(\tau)) \\
& - \frac{Q_0}{k} (a-r) \sin m\phi_0(\tau) P_n^m(\cos \theta_0(\tau))
\end{aligned}$$

where $n+m$ must be even for the summations

where

$$\hat{H}_{0\cos}(\lambda_{nl}, \gamma_n, m, \tau) = \frac{Q_0}{k} \int_0^\tau \left\{ \int_a^b \left[\frac{[[2-n(n+1)]r + an(n+1)] P_n(\cos \theta_0(\tau))}{\alpha} + \frac{a-r}{d\tau} \frac{d \cos \theta_0(\tau)}{d \cos \theta} \frac{d P_n^m(\cos \theta_0(\tau))}{d \cos \theta} r^2 \right] R_{nl}(r) dr \right\} e^{-\alpha \lambda_{nl}^2 (\tau-t)} d\tau$$

$$\begin{aligned}
\hat{H}_{\cos}(\lambda_{nl}, \gamma_n, m, \tau) = & \frac{Q_0}{k} \int_0^\tau \left\{ \int_a^b \left[\frac{[[2-n(n+1)]r + an(n+1)] \cos m\phi_0(\tau) P_n^m(\cos \theta_0(\tau))}{\alpha} \right. \right. \\
& + \frac{a-r}{d\tau} \cos m\phi_0(\tau) \frac{d \cos \theta_0(\tau)}{d \cos \theta} \frac{d P_n^m(\cos \theta_0(\tau))}{d \cos \theta} r^2 \\
& \left. \left. - \frac{a-r}{\alpha} m \frac{d \phi_0(\tau)}{d \tau} \sin m\phi_0(\tau) P_n^m(\cos \theta_0(\tau)) \right] R_{nl}(r) dr \right\} e^{-\alpha \lambda_{nl}^2 (\tau-t)} d\tau
\end{aligned}$$

$$\begin{aligned}
\hat{H}_{\sin}(\lambda_{nl}, \gamma_n, m, \tau) = & \frac{Q_0}{k} \int_0^\tau \left\{ \int_a^b \left[\frac{[[2-n(n+1)]r + an(n+1)] \sin m\phi_0(\tau) P_n^m(\cos \theta_0(\tau))}{\alpha} \right. \right. \\
& + \frac{a-r}{d\tau} \sin m\phi_0(\tau) \frac{d \cos \theta_0(\tau)}{d \cos \theta} \frac{d P_n^m(\cos \theta_0(\tau))}{d \cos \theta} r^2 \\
& \left. \left. + \frac{a-r}{\alpha} m \frac{d \phi_0(\tau)}{d \tau} \cos m\phi_0(\tau) P_n^m(\cos \theta_0(\tau)) \right] R_{nl}(r) dr \right\} e^{-\alpha \lambda_{nl}^2 (\tau-t)} d\tau
\end{aligned}$$

$$R_{nl}(r) = r^{-1} \left[J_{n+\frac{1}{2}}(\lambda_{nl}a) Y_{n+\frac{1}{2}}(\lambda_{nl}r) - Y_{n+\frac{1}{2}}(\lambda_{nl}a) J_{n+\frac{1}{2}}(\lambda_{nl}r) \right]$$

$$[R_{nl}(r), R_{nl}(r)] = \int_a^b r^2 R_{nl}^2(r) dr$$

This solution is being programmed on a workstation and the results will be presented at the conference.

SUMMARY

The temperature distribution in a hemispherical shell during a machining operation has been determined. A set of representative results will be available at the conference. Subsequently, the thermal stress distribution in the shell may be calculated using the temperature distribution as input.

Thrust Force and Wall Roughness in Microdrilling

Craig Friedrich, Yimin She, Ravi Muthu, Chirayu Dave

Institute for Micromanufacturing

Louisiana Tech University

Ruston, LA 71272

Abstract

Microdrilling is a process capable of creating deep, straight holes with good surface finish and close tolerances in a variety of materials. Microdrilling can also be used to perform post-processing of microcomponents created by other microfabrication techniques such as lithography and electroplating. In order to evaluate its usefulness, it was necessary to quantify the process so that proper fixtures could be designed to withstand the drilling thrust force and to evaluate the expected wall roughness of the drilled hole. It was found that the thrust force agreed well with preliminary comparisons of a three-dimensional microdrilling model currently under development using macroscale cutting mechanics with consideration to the thin-chip deviation of the specific cutting energy. The wall roughness was found to be very closely dependent upon the drill feed per revolution into the work material which closely correlates with the form of the macroscale equations for surface finish in turning but the drilled holes were extremely smooth with roughness of less than 100 nanometers rms attained.

Introduction

The Complementary Micromachining Processes Laboratory (*CMPL*) of the Institute for Micromanufacturing (*IfM*) has been using the micro-spade drilling process to fabricate holes 25 micrometers, and larger, in diameter in a variety of materials, such as plastics, aluminum, copper, and wrought nickel. These studies are in support of direct microcomponent fabrication and for rapid prototyping of masks and molds for general micromanufacturing. Microcomponents must be rigidly held during drilling however the microcomponents are obviously delicate and should not be exposed to higher fixture forces than necessary.

The microdrilling process uses a spade-type drill made of either M42 cobalt-high speed steel or C-2 micrograin tungsten carbide. The drills are commercially available and typically have tolerances of $+0$ and -2.5 micrometers. The length-to-diameter ratio of the drills varies from 4 for the smallest drills up to 14 for much larger ones. Because the microdrills do not have spiral flutes like a twist drill, there is typically a problem of chip removal from within the hole during drilling. Therefore, microdrilling is normally done with a peck drilling cycle wherein the drill is partially or completely removed from the hole between advances in depth. The advance in depth is termed the peck depth.

The drilling tests were conducted on a modified National Jet Co. 7-M manual drilling machine. The manual machine was modified so the drill feed (spindle axis) and one axis of work piece movement were computer controlled through a stepper motor and precision lead screw. The feed axis was controlled to provide a range of peck depths, feed per revolution of the drill, and total hole depth. The thrust force was sensed with a PCB Piezotronics force sensor mounted in line with the drill spindle. The thrust force event duration typically lasted 100 to 150 milliseconds which caused DC drift in the piezoelectric sensor. Therefore, a calibration error correction factor had to be determined. This was accomplished by using a spring with a known spring constant over a small deflection (a few millimeters). Dead weights were applied to the spring and the deflection was measured with an LVDT. The spring was found to be very linear. The force sensor was then placed so that one end of the spring was constrained against the sensing head. A computer controlled stepper motor and lead screw provided the spring deflection as a function of time and the resulting voltage output from the load sensor was recorded on a digital oscilloscope. Because the spring deflection, and therefore true load, was accurately known

as a function of time, the true load on the sensor and the sensor output voltage were compared as a function of time and a correction factor was obtained. The error calibration is shown in Fig.1. The calibration was very linear beyond 25 milliseconds duration and the multiplier was one for very short events (high frequency). This scalar correction was applied to all drilling thrust force data.

Thrust Force Results

The thrust force measurements were made while drilling 6061-T6 aluminum and nickel 200. In each case, five holes were drilled under the same set of parameters and the thrust force was measured at each peck. The parameters which were varied were drill rotational speed (4000, 6000, 8000 and 10000 rpm), drill feed (0.4, 0.6, and 1.0 micrometers per revolution), and peck depth (10, 20, and 30 micrometers). The results were both expected and unexpected. The thrust force varied relatively linear with feed per revolution in both materials, as expected. Thrust force was mostly independent of rotational speed in both materials, as expected. A very small sample of data collected is shown in Fig.2. The thrust force increases non-linearly from zero hole depth to the depth where the drill point is fully immersed in the work piece. For the 75 micrometer drill used to collect the data shown, the immersion depth is about 30 micrometers. Beyond this depth, the thrust force is nearly linear except for a slight increase due to increased friction between the drill and the hole wall and increased chips remaining in the hole. The larger force at 6000 rpm is a misleading artifact of the experimental procedure. These holes were drilled near the end of the procedure using the same drill throughout. The increased thrust force is because of increased drill wear. In fact, thrust force monitoring is often used in microdrilling to signal drill wear and required drill changeout.

The thrust force for a 30 micrometer peck was greater than for a 10 because of the larger amount of machined chips accumulating in the hole. This was particularly true for the nickel because of the fact that the nickel chips deposited onto the high speed steel drill because of magnetic attraction. This, compounded with the work hardening of the nickel, made for very high thrust forces. In fact, the thrust forces were so high in the nickel, that the tests used to gather the data shown in Fig.3, resulted in three broken drills. Because total hole depth is a function of drilling time, the shape of the thrust force data for the nickel looks very similar to a wear-vs-time plot for a typical cutting tool.

Surface Finish Results

The hole wall roughness was measured for both polymethyl methacrylate (often used for microformed structures and lithography masks), and UNS C11000 copper. The copper was used to compare the wall roughness of a microdrilled hole with the inside wall of the bore of an electroplated copper gear. The same three drilling parameters were varied and 10 holes were drilled for each combination. The 100 micrometer diameter holes were then sectioned and a WYKO RST was used to measure the roughness with 2-D linescans oriented along the hole axis and at five different circumferential locations. Therefore, the roughness data consisted of 50 data points for each set of drilling parameters. It was found that both feed per revolution and peck depth had the most influence on wall smoothness with peck depth being slightly more important. Feed certainly has a strong influence on surface finish in most all machining operations, and that is certainly true in microdrilling. However, in peck drilling, because the drill is removed at each peck cycle, the drill will repeatedly burnish the hole wall resulting in a very smooth surface. Using a smaller peck depth, that is a larger number of peck cycles to create a hole of constant depth, results in more burnishing and a smoother wall. A very brief summary of the extensive data collected is shown in Fig.4. Here, roughness is plotted against volumetric removal rate. For each removal rate, the uppermost point (roughest) of the three represents a 30 micrometer peck and the lowest (smoothest) point, a 10 micrometer peck. Generally, the higher removal rate resulted in a rougher wall, however it is possible to select a combination of speed, feed, and peck depth which can give a very smooth wall.

An electroplated copper gear was sectioned and the roughness of the inside surface of the central bore was measured. The bore was about 100 micrometers in diameter and five, 2-D linescans were made along the axis of the bore. The average rms roughness of the gear was 440 nanometers (+/- 85nm at 95% confidence). Ten, 100 micrometer diameter holes were drilled in the copper using the drilling combination which gave the smoothest wall in the prior studies. The parameters were a speed of 4000 rpm, a feed of 0.3 micrometers per revolution, and a peck depth of 10 micrometers. The holes were sectioned and the rms roughness of the drilled hole was 145 nanometers (+/- 24nm at 95% confidence).

Summary

It was found that the thrust forces in drilling 6061-T6 aluminum are in the range from zero to approximately 4 Newtons and as high as 25 Newtons for drilling nickel 200. In addition, many drills were broken while obtaining the data for nickel but only one drill was broken while drilling a total of 130 holes in the aluminum. This data will prove useful for designing and properly constraining microcomponents and materials for microdrilling. It was found that the wall roughness of microdrilled holes can be easily controlled by proper specification of the drilling parameters. A smoother wall will be obtained with a low feed and small depth per peck. Very recent studies have shown that the hole-to-hole repeatability of the drilled diameter can be limited to less than +/- 1 micrometer.

Acknowledgments

The authors acknowledge Ms. Tonya Gamblin, Mr. Mark Crum, and Mr. Marc Hayes, who performed the force transducer error correction factor studies.

Force Transducer Error Calibration

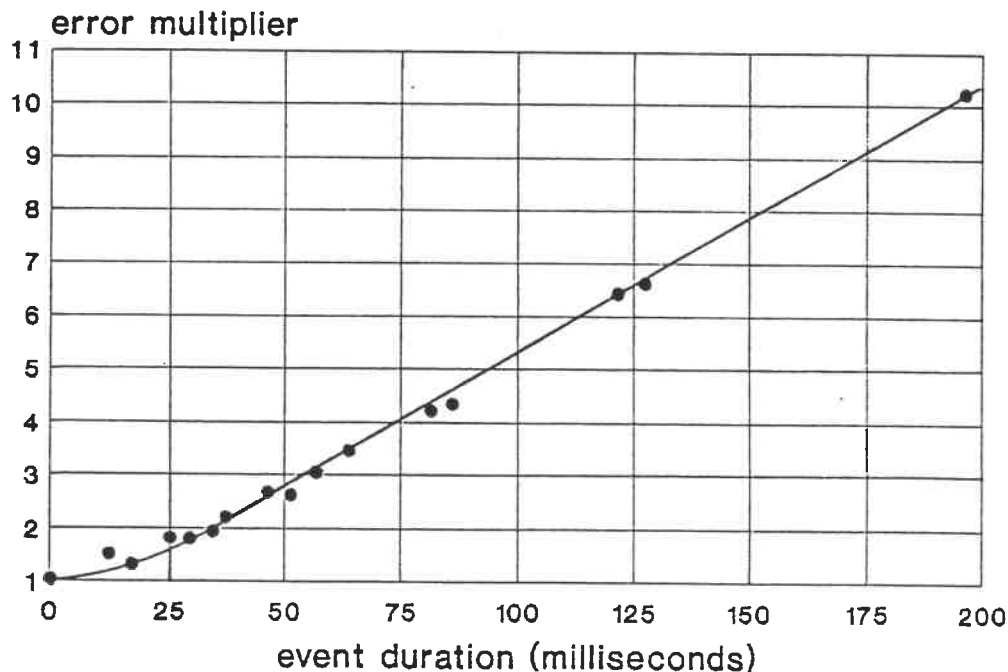


Fig.1. Force transducer error calibration

Thrust Force vs Feed

0.6 micrometers per revolution

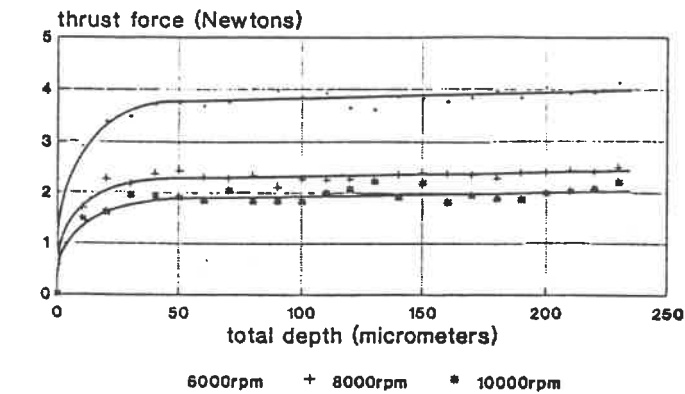


Fig.2. Typical thrust force data for aluminum

Thrust Force - Al and Nickel

0.6 micrometers/rev - 8000 rpm

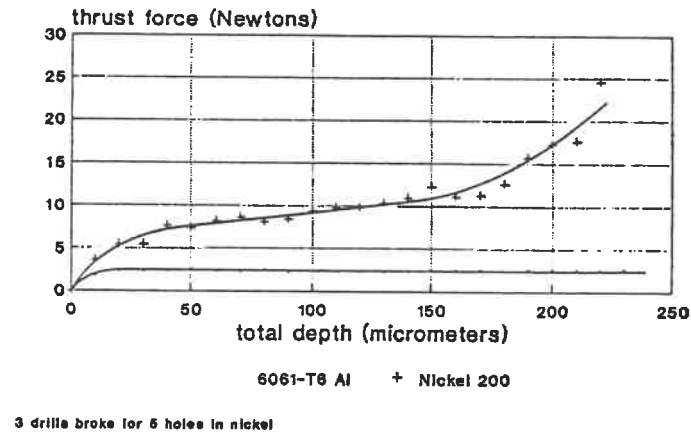


Fig.3. Thrust force comparison for aluminum and nickel

RMS Roughness vs Removal Rate

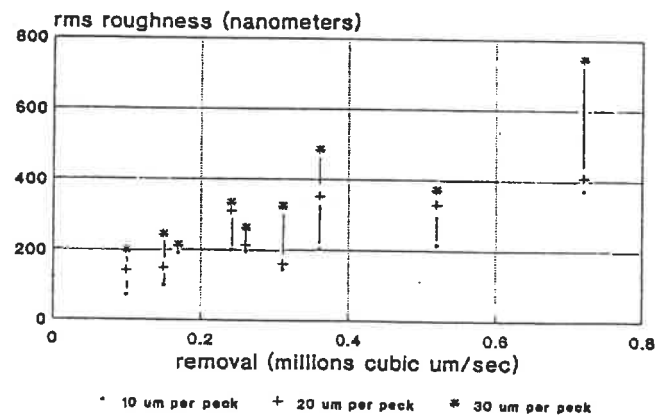


Fig.4. RMS roughness versus volumetric removal rate in PMMA

Mechanical Micromilling of PMMA

Craig Friedrich, Bharath Kikkeri, Thamyliniane Nagarajan
Institute for Micromanufacturing
Louisiana Tech University
Ruston, LA 71272

Abstract

At the macroscale, the milling process is very versatile and capable of creating three dimensional features and structures. Adaptation of this process at the microscale could lead to the rapid and direct fabrication of micromolds and masks to aid in the development of microcomponents. This task has been undertaken and results of the process indicate it can become an increasingly useful method. The micromilling process is currently characterized by milling tools with a diameter as small as 22 micrometers. The micromilling process can create trench-like features with vertical sidewalls and good smoothness. External corners are sharp and stepped features can be machined simply by programming those shapes.

Introduction

The Complementary Micromachining Processes Laboratory (*CMPL*) of the Institute for Micromanufacturing (*IfM*) has been developing the micromilling process to provide a method for the rapid creation of microfeatures in polymethyl methacrylate (PMMA) and other materials by direct material removal. The microfeatures may be shaped depressions in the PMMA which can be plated with suitable materials for the direct fabrication of components or the trench features may be plated to directly form a lithography mask. As with any process utilizing a physical cutting tool, the micromilling process can not create sharp interior corners because of the finite radius of the milling tool. Micromilling has not at present created a depth greater than 100 micrometers (although a depth of 1mm is currently being planned), and the process has not yet created walls less than 4 micrometers wide because of the presence of side cutting forces and deburring after machining. The micromilling process can, however, create trench structures in minutes to hours with very vertical sidewalls, good smoothness, sharp exterior corners, and relatively complex geometries. At present, the process appears to be well suited to complement lithographic processes.

X-ray Mask and Mold Making Process

X-ray lithography is conceptually similar to optical or ultraviolet lithography. The key differences are that the incident energy is a beam of collimated x-rays usually produced by a synchrotron, the mask substrate and absorber must be suitable for the efficient transmission and absorption of x-rays, and the photoresist is normally polymer-based with PMMA being widely used. Depending upon the energy of the x-rays, the thickness of the mask absorber (normally gold) must be on the order of 10 to 20 micrometers, with a thicker absorber required for very hard x-rays. The gold absorber is normally electroplated into trench-like structures in a resist material. In order to expose the resist to this depth requires either a high-energy electron beam writing instrument (and this tends to push the depth capability of such a tool), or a multi-step process whereby a chrome optical mask is first created by low energy e-beam lithography. This optical mask is then used to create an intermediate x-ray mask which has a thin gold absorber about 2 micrometers thick. The intermediate x-ray mask is then used in an x-ray lithography process to create the final x-ray mask which is 10 to 20 micrometers thick.

Because these are serial processes, the error budget required for the final parts or structures dictates that each step in the mask making process must have strict tolerances. Depending on the tolerances required for the final parts, the tolerances on each of the serial processes may result in a very

expensive and time consuming prototype development process. Certainly the lithography processes have advantages over direct removal processes such as structure placement accuracy, repeatability, mask-to-mask overlay accuracy, and feature size control. During initial product development, however, these advantages can often be sacrificed for rapid and low-cost results. Because of this, the mechanical micromilling process has been developed.

Micromilling Tool Fabrication

Micromilling tools are made with the focused ion beam machining process. The tool begins as a cylindrical blank of M-42 cobalt-high speed steel or C-2 micrograin tungsten carbide. The blank is 22 micrometers in diameter and has been ground from a diameter of 1.02mm. Material is then removed by impact from 20keV gallium ions in a controlled process. This is shown in **Fig.1**. The present tools have a geometry as shown in **Fig.2**. The tools have two very sharp cutting edges with a radius of 50nm to 100nm and two large macroscale rake faces to help eliminate work material adhesion. Because the depth of cut per cutting edge is between 50nm and 100nm, the macroscale rake angle is irrelevant. The tools and cutting conditions have been designed to preclude failure from torsional fatigue which is the primary in-service failure mode.

Micromilling Results

Micromilling is performed on a custom-built high precision micromilling/microdrilling/micro-EDM machine as shown in **Fig.3**. The machine was a joint project of the *IfM*, National Jet Co., and Dover Instruments. The 2.29mm diameter milling tool mandrel is rotated in a double vee-block with four convex diamond bearing surfaces. The tool is presently driven at 19,000rpm. Although the tool could be driven to 40,000rpm, results indicate this is probably not necessary.

To demonstrate the process, a test pattern was micromilled. The pattern is shown in **Fig.4** and consisted of three exponential spirals with a straight radial section. The spiral portion contains both straight and stepped-wall structures. The spirals were created so that the total angle was 1080 degrees and was incremented in 5-degree increments. The feedrate was 35 micrometers per second, the axial depth per pass was 2 micrometers, and the total depth was 62 micrometers. The time to mill the pattern was about three hours and well over 35 million cubic micrometers of material was removed.

A portion of the stepped-wall region is shown in **Fig.5**. It can be observed that the width of the step structures exponentially decays from top to bottom thus further demonstrating the versatility of the process. A view along the straight radial trench is shown in **Fig.6**. Roughness measurements made with the WYKO RST indicated the mean bottom roughness was 65-90nm. Also, the process can create sharp exterior corners with a radius of 2-3 micrometers. The radius of interior corners is of course limited by the radius of the milling tool. The sidewalls form an angle of 89.5 degrees with the trench bottom. It was learned after machining that the milling tool had 0.44 degrees of front taper so the tool geometry is reproduced very faithfully in the work piece.

The spirals also allowed study of the minimum wall thickness which was attained. The upper portion of **Fig.7** shows a wall section which is 8 micrometers wide by 62 micrometers deep. Because the work is deburred in a solution of 1:2 acetone in ethanol, the agitation broke walls thinner than this. However, in other milling work, walls 4 micrometers wide separating 10 micrometer-deep trenches were machined.

Summary

The micromilling process has been developed to the stage where it appears useful for fabricating molds and mask structures in PMMA. The process integrates focused ion beam micromachining for fabrication of the milling tools and utilizes a high precision milling machine designed for the milling process. To date, a milling tool has not broken during machining when used in accordance with the machining parameters presented. Tool failure did not occur while machining with a single axial depth of cut of 28 micrometers and a total length of 8mm, however the tool failed due to obvious fatigue shortly

thereafter. The longevity of the tools indicates the process can be cost effective, especially where the application is for x-ray mask making. Trenches as deep as 62 micrometers have been milled in PMMA, however for a typical x-ray mask the depth would need to be on the order of 10 micrometers. The next step in the process development is to actually fabricate and test an x-ray mask made by micromilling.

Acknowledgment

The authors wish to thank the Air Force Office of Scientific Research for partial funding for the development of the high precision milling machine under grant F49620-93-1-0547.

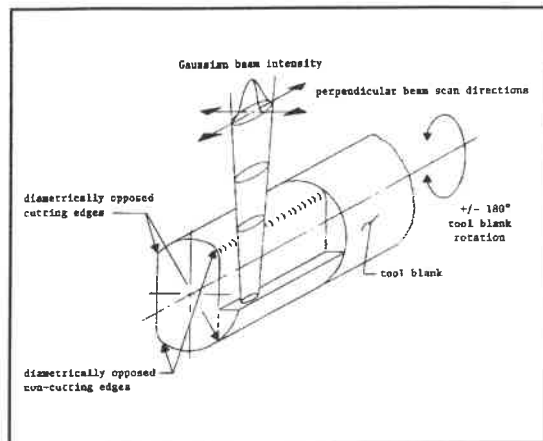


Fig.1 Micromilling tool fabrication

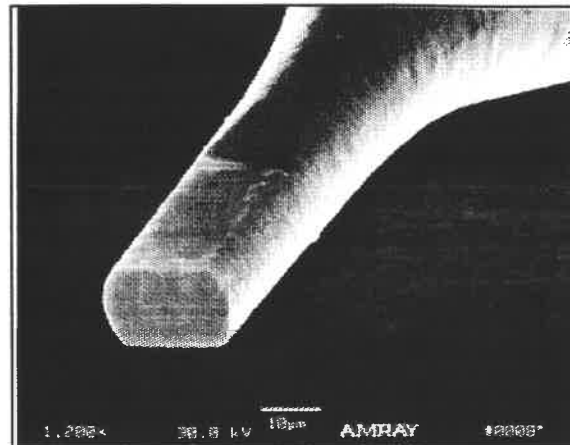


Fig.2 2-fluted milling tool in high speed steel
(scale bar = 10 microns)

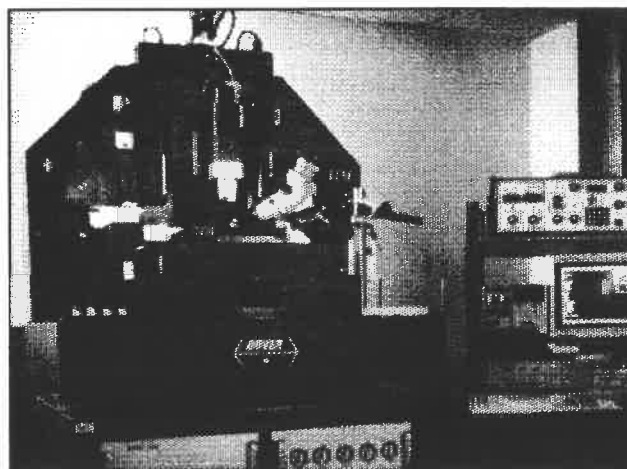


Fig.3 High precision micromachining tool

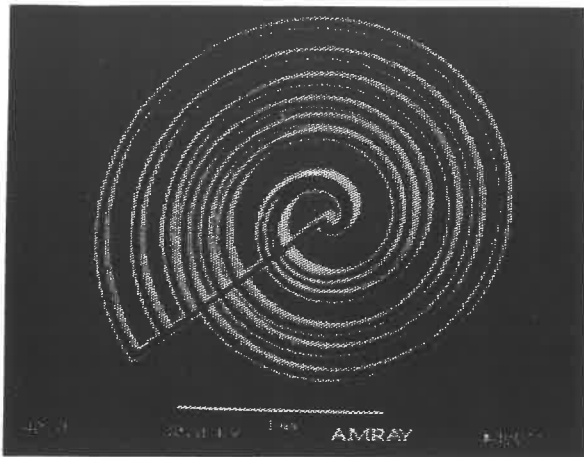


Fig. 4 Spiral pattern (scale bar = 1 mm)

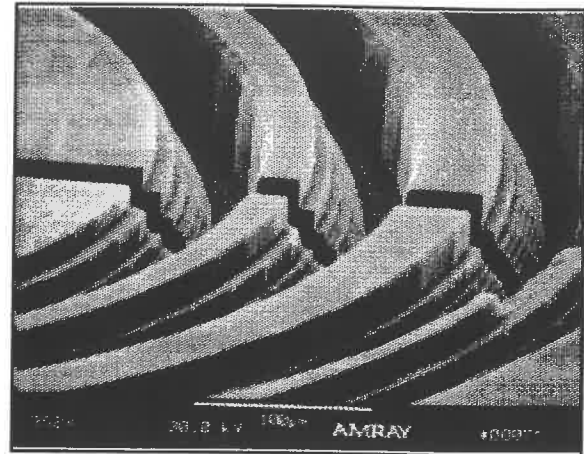


Fig. 5 Straight and stepped walls (scale bar = 100 micrometers)

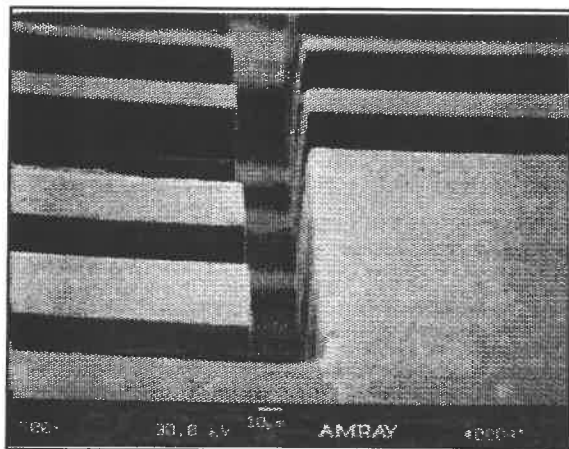


Fig. 6 Radial trench toward pattern center (scale bar = 10 micrometers)

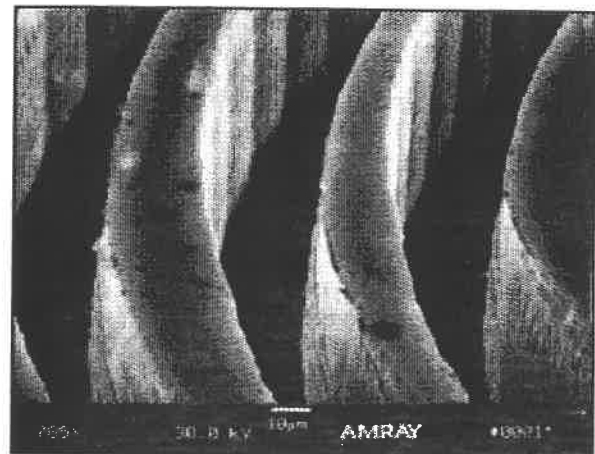


Fig. 7 8-micrometer wide wall (scale bar = 10 micrometers)

A PRELIMINARY STUDY ON THE EFFECT OF TOOL FLANK WEAR ON SURFACE FINISH IN END MILLING

Shreyes N. Melkote

The George W. Woodruff School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Georgia

1 Introduction

Machined surface finish is complex in that it contains a wide range of spatial frequencies. The origin of only a few of the spatial frequencies can be accounted for easily based on the cutting tool geometry and process kinematics. Other important sources that contribute to the surface texture include tool-workpiece vibration, chip formation/rupture, built-up edge, and tool wear. The effect of tool-workpiece vibration on the surface roughness has been investigated by other researchers while the contribution of the chip formation process is not sufficiently understood to allow model development. Although it is clear that the gross effect of tool flank wear is to degrade the surface finish as machining proceeds, very few studies -- experimental or theoretical -- have been reported in the literature that throw light on the mechanism by which the surface finish deteriorates. This is especially true for the peripheral end milling process.

This paper presents the results of a preliminary experimental study to understand and model the geometric effect of tool flank wear on the surface roughness produced in peripheral end milling. Such a model will have potential applications in planning tool change operations and controlling the effect of tool flank wear on the surface finish. Both artificial and actual end mill wear tests are carried out and the effects of different flank wear levels on the surface finish are studied. In addition, a simple but commonly assumed geometric model for flank wear is used to evaluate its effectiveness in simulating the experimental observations.

2 Experimental details and results

2.1 Artificial tool wear tests. The effect of tool flank wear on surface finish produced in up- and down-peripheral end milling was studied by artificially grinding six different levels of flank wear on indexable uncoated carbide inserts (Kennametal K-1, square, 8.3 mm size; 19.05 mm dia. end mill, 0° helix and radial rake angles). Only one insert was used to avoid the influence of runout. A machinable wax workpiece was used to minimize the effects of machining dynamics on surface finish and to highlight only the influence of tool geometry, kinematics and flank wear. Although the wax workpiece is susceptible to some damage by the stylus-tip during surface assessment, it yields considerable insight into the surface generation process in the presence of tool flank wear. Straight peripheral milling cuts (0.508 mm/rev feed, 600 rpm, 7.62 mm axial depth, 0.381 mm radial depth) were made with each worn insert on a 2.5 axis Bridgeport milling machine. The surface finish was measured by a single-skid stylus profilometer (Perthen-Mahr Model C3A; 4 μm sampling interval) connected to a data acquisition computer. No electronic surface filtering was performed.

The effect of flank wear on the amplitude and spacing of the surface peaks and valleys in the surface profiles is shown in Figure 1(a)-(e). The mechanical interaction between the worn tool and the machined surface gradually obliterates the tool feed marks. For a wear land size of $W_i = 0$ the roughness peaks are, as expected, uniformly spaced at the feed per revolution. For $W_i = 0.165$ mm the roughness spacing is no longer very uniform. The roughness height increases significantly for $W_i = 0.381$ mm and 0.508 mm and the feed marks become indistinguishable. The power spectral density plots shown in Figure 1(d)-(f) confirm this trend. For the sharp tool case the power is concentrated at the feed spatial frequency (0.00196 μm^{-1}). As the tool wears, the power contribution of the feed spatial frequency reduces while a smaller spatial frequency component (4.88×10^{-4} μm^{-1}) emerges and increases in magnitude. However, the

physical origin of this lower spatial frequency is not easily explained. It is possible that this frequency is related to errors in the feed mechanism of the milling machine.

The surface profile/power spectral density results obtained in up-milling are similar to those for down-milling except that the surface degradation takes place much quicker in up-milling. This is most likely due to the longer tooth path encountered in up-milling which results in a longer chip that often gets entangled and re-cut during subsequent revolutions of the tool. The variation of surface roughness parameters (R_q , R_{max}) with flank wear in up- and down-milling are shown in Figures 2(a)-(b). In both milling modes there is initially a slight decrease in the roughness values before increasing significantly. Similar trends have been reported in turning (Allen & Brewer, *Adv. Mach. Tool Des. Res.*, 1965; Solaja, MR73-242, SME).

2.2 High speed steel tool wear tests. Uncoated high speed steel (HSS) tool wear tests were conducted with a single flute helical end mill (19.05 mm dia., 30° helix, 75 mm length) on a Cincinnati Milacron 7VC vertical machining center. The machining conditions were: 0.508 mm/rev feed, 300 rpm, 17.78 mm axial depth, 0.762 mm radial depth, 152.4 mm length of cut per tool pass. A 152.4 mm x 76.2 mm x 25.4 mm rectangular block of AISI 1045 steel (197 BHN) was rigidly bolted to a three-component cutting force dynamometer (Kistler Model 9257A; 1600 Hz per channel sampling frequency). Up-milling tests were run to minimize the tool vibration due to tool impact upon engagement. Under these conditions it is easier to distinguish between the effects of tool wear and the tool-workpiece vibrations on surface finish. The helical end mill exhibits a distinct variation in the width of the flank wear land along the cutting edge. In view of this, flank wear and surface profile measurements were made at two locations (measured from the free end) along the axis of the end mill locations close to the leading (5.08 mm) and trailing portion (15.24 mm) of the tool cutting edge. The total length of workpiece material machined was 13.72 m corresponding to 90 tool passes.

The progress of HSS tool flank wear at the leading (5.08 mm) and trailing (15.24 mm) edge locations is shown in Figure 3. It is clear from the figure that overall the trailing edge experiences greater flank wear than the leading edge. An examination of the power spectra of the measured surface profiles in Figure 4 shows the same general trend as seen in the artificial tool wear tests. The trend however is weaker than in the artificial tool wear tests. This is partly because the test was stopped at a flank wear level of only 0.1778 mm due to chipping of the tool.

3 Flank wear geometry model

The validity of a simple but commonly assumed flank wear geometry model (Ismail, et al, ASME J. Eng. Ind., 1993) (see Figure 5) was examined for the purpose of surface finish prediction. This model is based on the macroscopic observation that tool wear causes a change in the radial position and shape of the cutting edge. The radial position of any point P along the wear land, R_P , and its angular position measured from the cutting tip (θ_P) can be calculated as follows:

$$R_P = \sqrt{R_W^2 + W_i^2} \quad (1)$$

$$\theta_P = \tan^{-1}(W_i / R_W) \quad (2)$$

The effect of flank wear on the 3-D surface topography generated in peripheral end milling can be simulated by combining eqs. (1) and (2) with the parametric tooth path equations describing the motion of the peripheral cutting edges. These equations include the effects of static tool deflection due to the cutting forces. They have been derived in detail elsewhere (Melkote, Ph.D. thesis, 1993) and are given below:

$$X_i = X_d(\theta_{rot}) + f_r |\theta_{rot}| / 2\pi + R_P \cos(\theta_i + u) \quad (3)$$

$$Y_i = Y_d(\theta_{rot}) + R_P \sin(\theta_i + u) \quad (4)$$

$$Z_i = R_i u / \tan \lambda \quad (5)$$

(X_i , Y_i , Z_i) are the coordinates of a point on the i^{th} end mill tooth; θ_{rot} is the cutter rotation angle measured positive counterclockwise from the X axis; θ_i is the angular position of the i^{th} tooth and is given by

$\theta_i = \theta_{rot} + (i - 1) * 2\pi / N_t + \theta_P$; $(X_d(\theta_{rot}), Y_d(\theta_{rot}))$ are the coordinates of the deflected center of the end mill at a particular instant and axial location; N_t is the number of end mill teeth; u is the wrap angle due to the helix (λ); R_t is the nominal radius of the end mill tooth.

The predicted and measured roughness parameters are shown in Figure 6. Increasing number of tool passes is synonymous with increasing flank wear magnitude. The predicted R_{max} values do show a change with increase in flank wear. However, the experimental trends are not captured very well in the predictions. The little variation that is predicted by the model can be attributed mainly to the X-Y displacements of the end mill and changes in the end mill radius due to tool wear. The agreement in R_{max} is not very good for the $Z = 15.24$ mm case. The R_q values match reasonably well but do not show much variation with flank wear. This is partly because the test was terminated early due to tool chipping.

There are several reasons for the observed differences between the flank wear model and the experimental results. Firstly, the model assumes the worn surface of the tool flank to be perfectly smooth and the heel of the wear land to be a sharp point. In reality, the worn region of the tool flank has its own roughness that interacts with the workpiece surface generated by the tip of the cutting tool. This interaction is not well understood at the present time to permit accurate modeling. Secondly, the cutting edge becomes rounded as the tool wears. This affects the chip formation process and the impact forces upon tool entry. Other factors influencing surface generation include material inhomogeneities, tool-workpiece vibration, etc. Work is in progress to address these issues.

4 Conclusions

This paper presented an experimental study into the geometric effect of tool flank wear on the surface finish produced in peripheral end milling operations. It was found that the contribution of the feed spatial frequency to the total power spectral density decreases with increase in the flank wear magnitude. Evidence of this trend is strongly present in the artificial wear tests. The actual HSS wear tests also indicate a similar trend. Also, the effect of tool flank wear on the surface generation cannot be simply described by the kinematic interaction between a smooth, flat wear land and the machined surface. The worn surface of the tool has a texture of its own that needs to be modeled also.

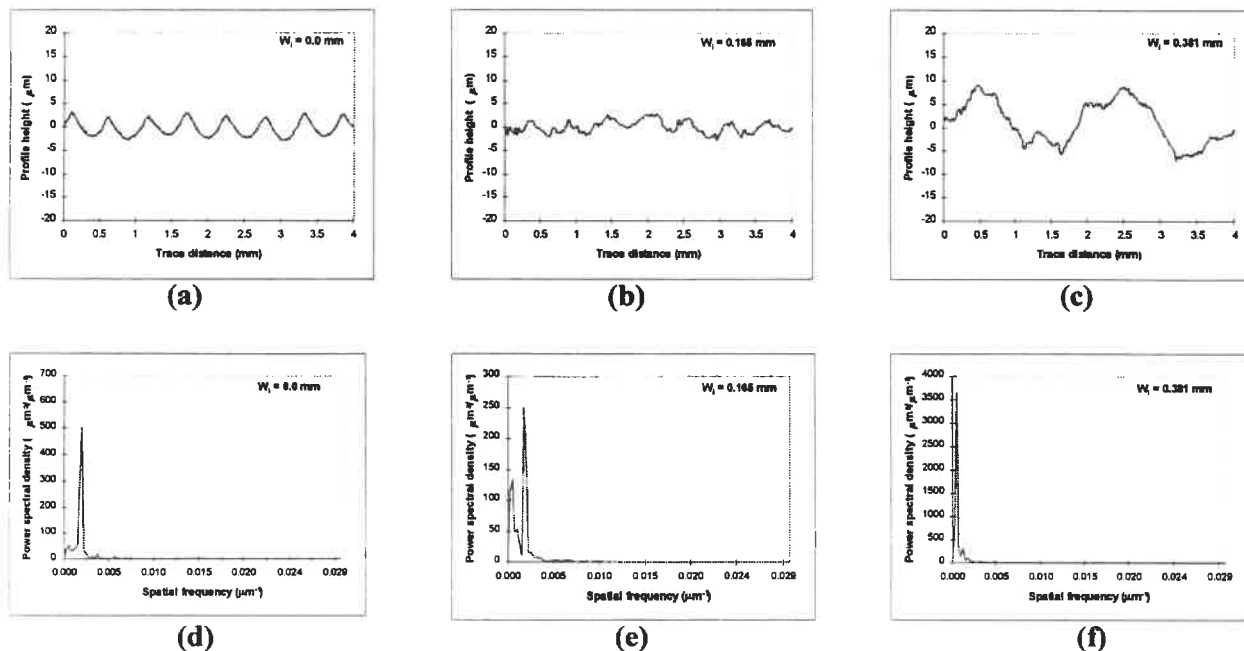
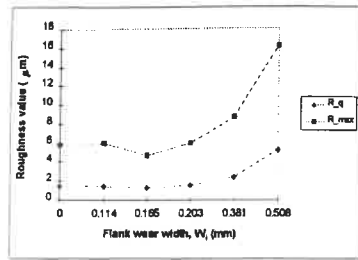
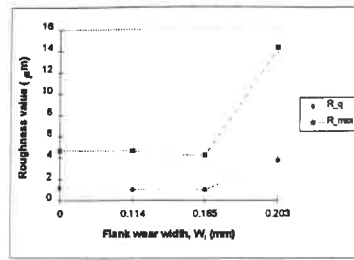


Figure 1 (a)-(c). Representative down-milled surface profiles at different levels of flank wear; **(d)-(f).** corresponding power spectral densities. Workpiece: machinable wax.



(a)



(b)

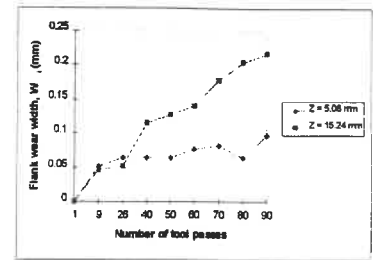
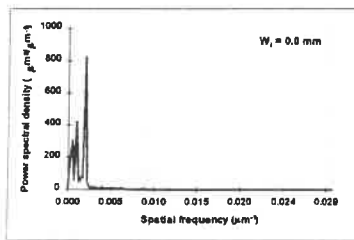
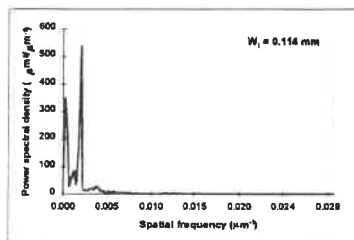


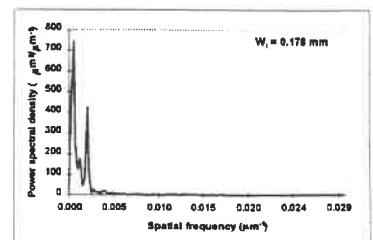
Figure 3. Flank wear growth in HSS tool wear test.



(a)



(b)



(c)

Figure 4 (a)-(c). Power spectral density for different flank wear levels of HSS tool.

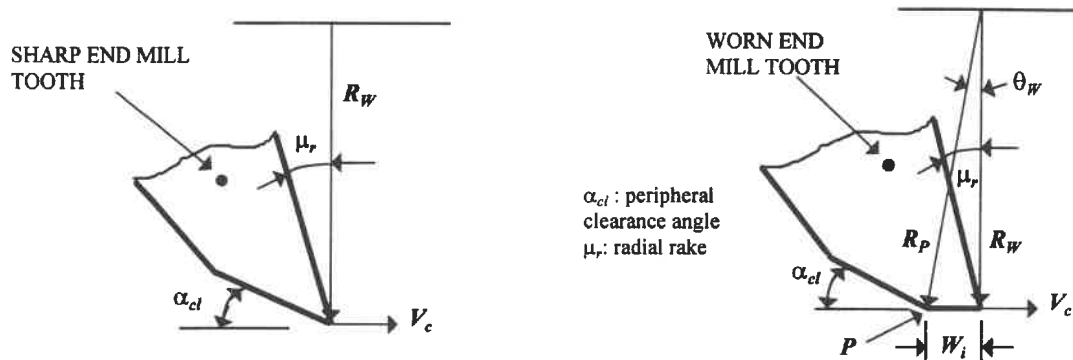
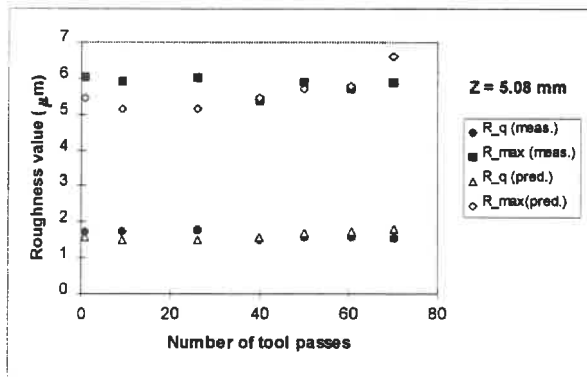
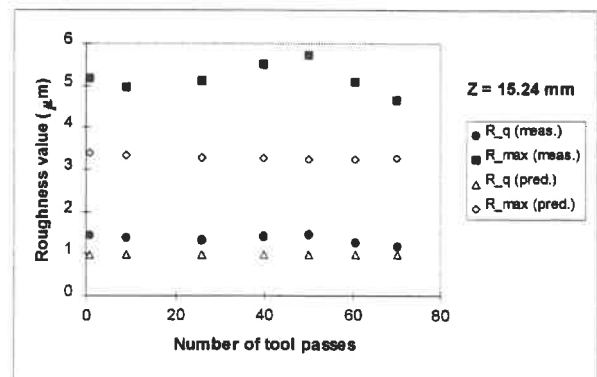


Figure 5. Flank wear geometry model.



(a)



(b)

Figure 6. Comparison of measured and predicted roughness parameters for different flank wear levels.

EXPERIMENTAL INVESTIGATIONS INTO ELECTRO-CHEMICAL SPARK MACHINING OF KEVLAR EPOXY COMPOSITES

V.K.Jain, S.K.Choudhury, V.Nesarikar,
Mechanical Engineering Department, Indian Institute of Technology,
Kanpur - 208016, INDIA

Present work deals with the machining of composite materials, namely, kevlar-epoxy composite, using a hybrid unconventional machining technique called **Electro-Chemical Spark Machining (ECSM)** process. This process uses electric discharge phenomenon which generates heat for the purpose of material removal; melting and vaporization being the mechanism of material removal. Since no mechanical force exists between the tool and the workpiece, hard and brittle materials like glass, ceramics and various composite materials can be conveniently machined by this process. Present work reports the findings about the effects of various process parameters on the responses obtained during slicing of kevlar-epoxy composites using a travelling-wire ECSM set-up.

PROCESS MECHANICS

The mechanics of the process can be explained as follows: A simple electro-chemical cell consists of two electrodes dipped in electrolyte. When an external potential is applied between the electrodes, electric current flows through the cell resulting in electrochemical reactions leading to anodic dissolution, cathodic deposition, electrolysis of electrolyte etc. If a suitable electrolyte is chosen and the electrodes are of grossly different sizes, then beyond a certain value of the applied potential, electric sparks appear at the smaller electrode and the cell current drops. This is known as electro-chemical discharge phenomenon. The electric discharge involves two steps: i) bubble generation during electro-chemical reactions as well as water vaporization and, ii) breakdown of the dielectric medium when the potential difference acting across the gas bubbles is greater than the breakdown voltage of the gas.

EXPERIMENTATION

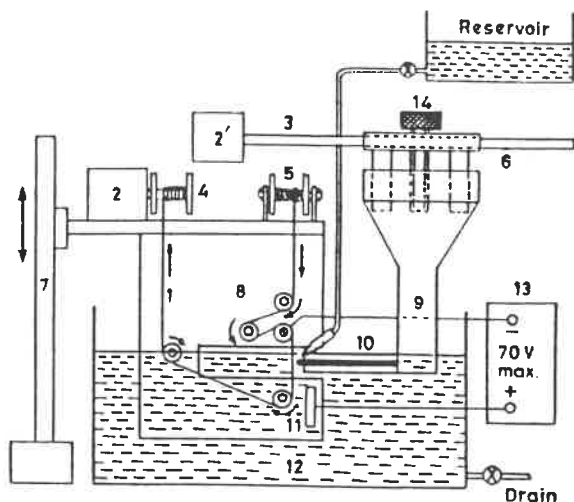
Schematic diagram of the experimental set-up is shown in Fig.1. Experiments have been conducted with sodium hydroxide as electrolyte. The electrolyte was flushed using an overhead reservoir for i) keeping temperature variations within the limits, ii) removing the graphite particles from the tank and minimizing a change in the specific conductance of the electrolyte and improving the visibility of the process by reducing the contamination of the electrolyte, and iii) removing the debris of workpiece from the electrolyte tank.

In the present work, experiments are carried out to study the effects of different variables (voltage, electrolyte concentration and specimen thickness) on the responses (material removal rate and diametral overcut) and are planned according to the statistical technique of experimental design.

Using the responses which are obtained experimentally [1], the following response surface equations are obtained:

1 Wire; 2, 2' Stepper motor; 3 Ball screw and slide; 4 Take-up spool; 5 Feed spool; 6 Slide block; 7 Sliding support for wire feed device; 8 Set of pulleys; 9 Joint rod; 10 Work piece holder; 11 Graphite anode; 12 Electrolyte tank; 13 Power supply; 14 Screw

Fig. 1 Schematic diagram of ECSM set up.



$$Y_{MRR} = 2.294 + 1.39X_1 - 0.00799X_2 + 0.5151X_3 + 0.3659X_1^2 - 0.3856X_2^2 + 0.3659X_3^2 - 0.06250X_1X_2 - 0.15X_1X_3 - 0.025X_2X_3$$

$$Y_{O_t} = 3.536 + 0.4002X_1 - 0.03936X_2 - 0.9053X_3 - 0.0922X_1^2 - 0.1306X_2^2 - 0.1783X_3^2 + 0.02575X_1X_2 - 0.0657X_1X_3 - 0.00925X_2X_3$$

where, X_1 represents voltage ranging from 46.5 volts to 63.5 volts, X_2 represents % concentration of electrolyte ranging from 11.5% to 28.5% and X_3 represents specimen thickness ranging from 2.7 mm to 9.82 mm.

Using these models, effect of different factors viz. voltage, electrolyte concentration and specimen thickness on the responses viz. MRR and diametral overcut (O_t) have been studied and discussed below.

RESULTS AND DISCUSSIONS

The effect of voltage variation on MRR for kevlar-epoxy composites is shown in Fig.2 for constant electrolyte concentration and varying specimen thickness where MRR is seen to increase with the increase in voltage. An increase in voltage implies higher discharge energy, and hence, more heat generated resulting in enhanced MRR. Moreover, with the increase in voltage the electrolysis process is accelerated, hence, the rate of generation of hydrogen gas bubbles is also increased and consequently, the rate of generation of discharge energy increases. These two factors together cause an increase in MRR with the increase in supply voltage.

MRR increases with an increase in electrolyte concentration up to 20% and then decreases (Fig.3). This is due to the fact that the specific conductance of NaOH electrolyte increases up to a range of 20 to 22.5% concentration and then starts decreasing. An increase in specific conductance means an increased electrolyte conductivity and consequently more electrolytic current. The process of electrolysis is accelerated by this fact which results in a greater rate of generation of H_2 gas bubbles at the cathode surface. Since the sparking takes place across the H_2 gas bubbles, an increase in H_2 gas bubble formation implies an increase in the rate of sparking and

hence higher MRR.

MRR decreases initially up to 3 to 4 mm thickness of the specimen and then increases with an increase in the specimen thickness (Fig.4). Falling nature of the curve may be attributed to the fact that the increased specimen thickness introduces extra obstruction to the hydrogen bubbles accumulating on the wire which is inside the groove being cut. This results in reduced sparking and hence MRR is decreased. Beyond this minima, MRR is found to increase consistently [1,2]. This can be explained by the fact that only a small part of the spark energy is being used in cutting the specimen and the rest is going as a waste in heating the electrolyte. Increased specimen thickness permits the cutting to take place along the larger length of the wire with almost the same energy generation per unit length per unit time resulting in higher MRR.

The dependence of overcut on voltage for different values of electrolyte concentration is depicted in Fig.5. Here O_t is seen to increase with an increase in the voltage. This can be explained by the fact that the discharge energy increases with an increase in the voltage and the electrolysis process is also accelerated causing an increase in the rate of H_2 gas bubble generation. These two factors together cause an increase in MRR and hence the overcut [3]. However, one of the important observations made during the experimentation is that the diametral overcut decreases with the increase in the specimen thickness as depicted in Fig.6 for varying electrolyte concentration. This phenomenon can be explained by the fact that an increase in the specimen thickness causes the H_2 gas bubble generation rate in the vicinity of workpiece to fall resulting in reduction in the discharge energy produced at the wire-workpiece interface decreasing the overcut. Hence it can be concluded from this observation that TW-ECSM is more advantageous for slicing of composites with higher thicknesses since qualitatively it improves the straightness of the machined component.

SURFACE ANALYSIS

The Scanning Electron Microscope photograph of a machined slot in the kevlar-epoxy composite specimen (Fig.7) shows that the cut surface obtained is not smooth but kevlar fibers protrude out of epoxy matrix resin. This is due to the fact that the kevlar and the epoxy have decomposition temperatures of $540^{\circ}C$ and $200^{\circ}C$ respectively. Due to this, for the same amount of time during cutting and for the same amount of heat input, amount of epoxy resin removed will be more than that of kevlar fibers.

REFERENCES

1. V.Nesarikar - "Travelling wire Electro-Chemical Spark Machining of Kevlar-Epoxy Composites", M.Tech.thesis, I.I.T.,Kanpur, July, 1994.
2. K.Allesu, M.K.Muju and A.Ghosh - "Experimental observations in the ECD machining of non-conducting materials" Proc of Int.Symp.for Electro Machining (ISEM-9),1989.
3. V.K.Jain, P.S.Rao, S.K.Choudhury and K.P.Rajurkar - "Experimental investigations into TW-ECSM of composites", Trans. ASME, Jour. of Engg.for Ind.,Vol.113,1991,pp.75

ACKNOWLEDGEMENT

Authors acknowledge the financial assistance from CSIR, New Delhi, for the project on "Sludge free ECSM of Advanced Engineering Materials" under which this work has been carried out.

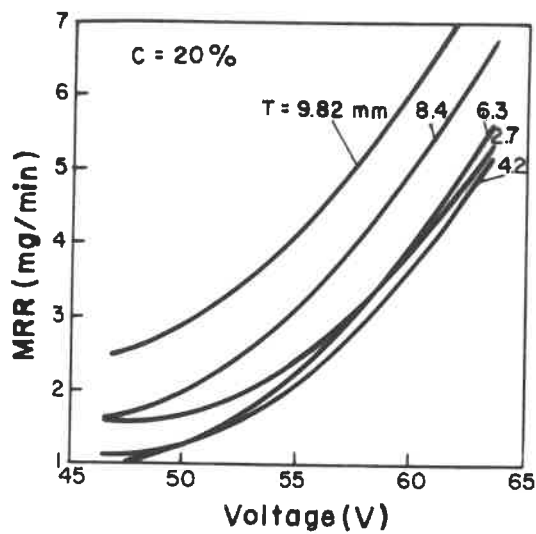


Fig.2. Effect of V on MRR

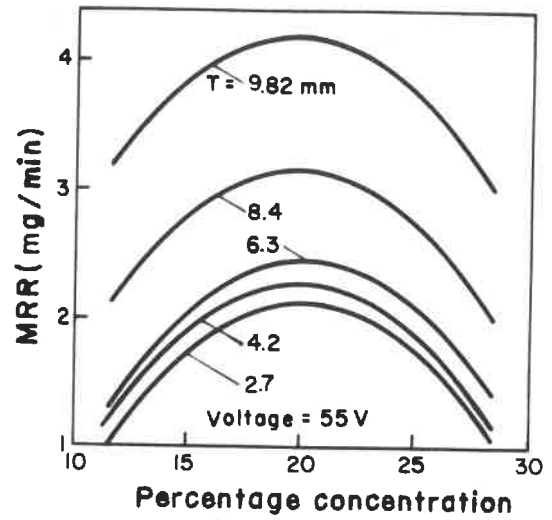


Fig.3. Effect of %C on MRR

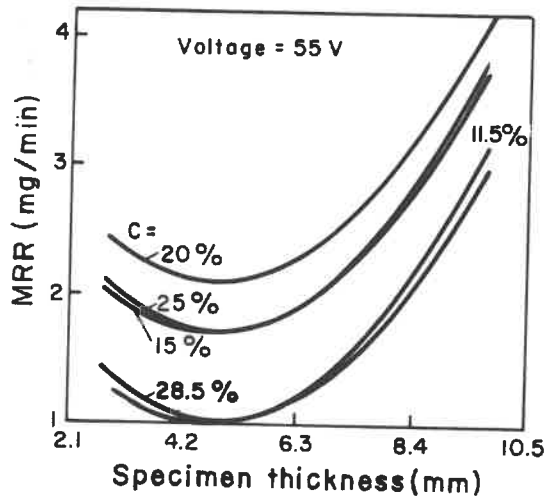


Fig.4. Effect of T on overcut

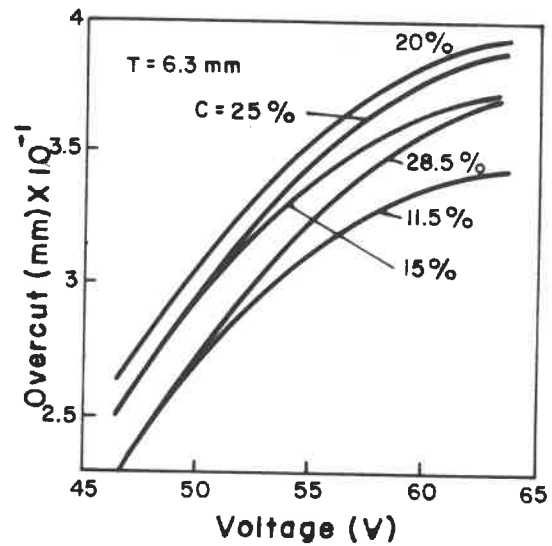


Fig.5. Effect of V on overcut

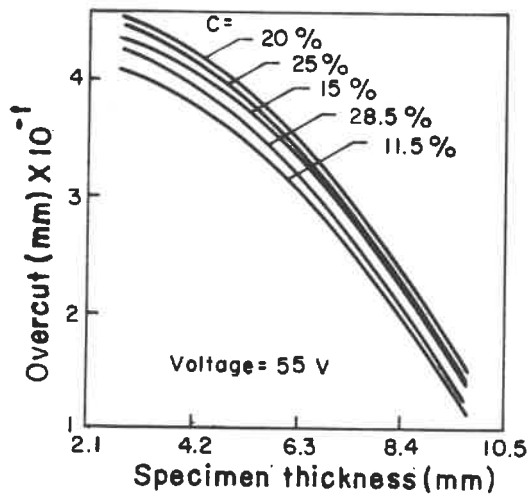


Fig.6. Effect of T on MRR

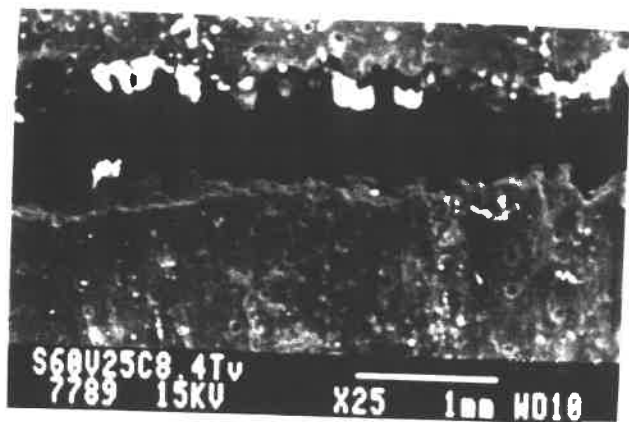


Fig.7. SEM photograph of the cut specimen

Cold Form Tapping of Internal Threads for Landing Craft in High Strength Steel

Ma Ping Wang Min

Department of Mechanical Engineering , Nanjing University of Aeronautics & Astronautics ,
Nanjing, Jiangsu Province, P.R. China 210016

Abstract

The fatigue strength of internal threads is the major factor influencing the life of the landing craft . Generally, the internal threads of the landing craft are shaped with cutting, then strengthened with special techniques, however the anti-fatigue life can not satisfy the requirements of the aircraft. In order to enhance the fatigue strength of the internal threads, this paper proposes a new anti-fatigue technology, cold form tapping of internal threads in high strength steel , to make the life of landing craft equal to that of the aircraft. Due to the high strength steel, the large diameter internal threads are much difficult to produce, and the general tapping process usually could not go smoothly while the success ratio is relatively low. To solve this puzzle, a particular supplemental equipment is developed, meanwhile an effective monitoring system is also established to improve the tapping quality. In this paper the structure of the special tap and features of the cold form tapping process are also discussed. Furthermore, the fatigue tests have been carried out, the results show that the fatigue life of vibrating tapping threads is five to thirty times than that of the cutting threads. In the end, the mechanism of the prolonging of fatigue lives of the special tapping formed threads are also analyzed .

Key words: Cold form tapping, internal threads, anti-fatigue, monitoring and control

Introduction

With the development of Aeronautical and Astronautical industry, the requirements for the quality of components are getting higher and higher. Especially, some important components such as the landing craft usually works at heavy and dynamic loads, the strong impact often leads to the failure of the parts. The investigates indicate that more than 90 percent of the failure on the landing gear is due to the damage of the internal threads in its joints, moreover, its cost is rather expensive. The improvement of the anti-fatigue life of internal threads in the landing craft is most important on the benefit of society and economic.

It is well known that internal threads is formerly produced by cutting, whose lower fatigue strength can not satisfy the requirements of design. In order to improve fatigue strength, most efforts have taken such as laser enhancing , spraying ball and so on. However, few effects have been achieved, especially for the 300M high strength steel, as it is sensitive to stress concentration.

This paper proposes a new anti-fatigue technology, cold form tapping of internal threads for landing craft in high strength steel [1]. Cold form tapping process has been paid attention nearly twenty years. Previous works were focused on the low-tensile high ductility material, such as aluminum alloys, copper alloys, low carbon steels and alloy steels. All maximum hardness values no more than 250HB, but no success have been reported on the high strength steel, it is because of its higher yield stress limit, vast elastic and plastic deformation easily lead to blocking and damage of the tap.

In this paper, a special attachment is developed, and some new technologies are introduced in section 2. Section 3 establishes a effective monitoring system. In section 4 ,the fatigue tests have been carried out and the mechanism on the prolonging of fatigue life is also investigated.

Experimental setup and discussion

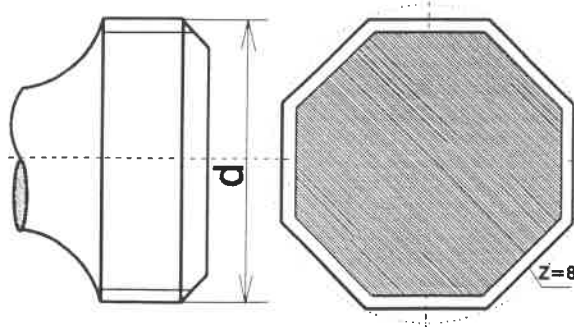


Fig.1 The structure of Taper M90*1.5

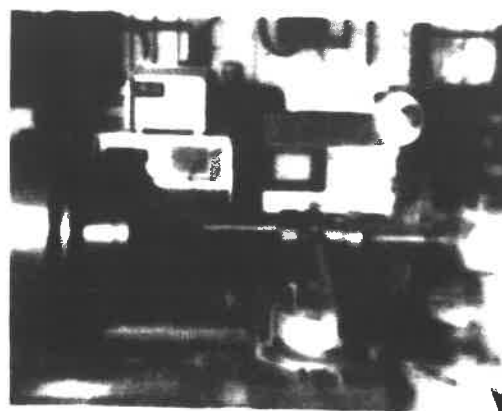


Fig.2 The experimental system

The standard to internal threads of the component in landing craft is M 90 × 1.5 with 35mm long, the diameter of the cored hole is designed as 89.15mm. In order to realize the cold form tapping of the internal threads, the particular taper is designed, which has eight lobe configuration arranged in a straight line along axis in high special steel shown in fig. 1, meanwhile, a special attachment has been developed to fit the conventional machine tool, as picture 2 including tool carriage, jig and monitoring & control system. A series of tests show that cold form tapping of threads easily produce blocking and its success ratio is no more than 10 percent. The maximum torque for normal deformation exceeds 1200N · m. To delimit the deforming energy, another technology, the vibrating tapping, plays key role in this process.

The kinematics of process is described as figure3.

$$\begin{aligned}
 a_t &= a \sin \omega t \\
 \theta &= \operatorname{tg} \frac{a_t}{b} \approx \frac{a_t}{b} \quad (a_t \ll b) \\
 \frac{d\theta}{dt} &= \frac{a}{b} \omega \cos \omega t \\
 \frac{d^2\theta}{dt^2} &= -\frac{a}{b} \omega^2 \sin \omega t \\
 M_v &= I_z \frac{d^2\theta}{dt^2} = \frac{1}{8} m d^2 \frac{a}{b} \omega^2 \sin \omega t \quad (1) \\
 \omega &= 2\pi f
 \end{aligned}$$

Where, M_v is the vibration torque, and θ the additional angle, $\frac{d\theta}{dt}$ the angular velocity, f the vibrating frequency, and a the amplitude of the vibrator.

The vibrating machining provides the taper a strong impact resulting in the transient elastic and lastic energy decreasing shown as figure 4. Analysis indicates that the regular contacting and departing between the taper workpiece produce pulselike discontinuous tapping effect which concentrates the tapping energy, decrease the deformation energy. In

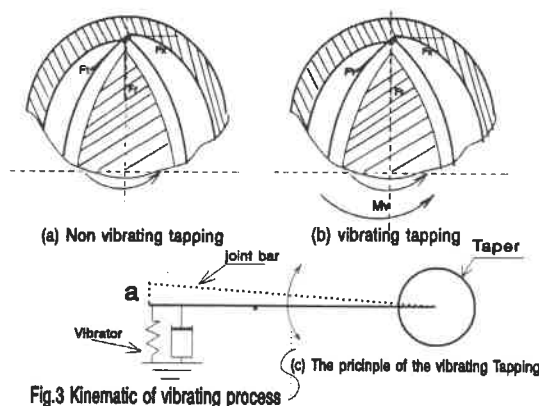


Fig.3 Kinematic of vibrating process

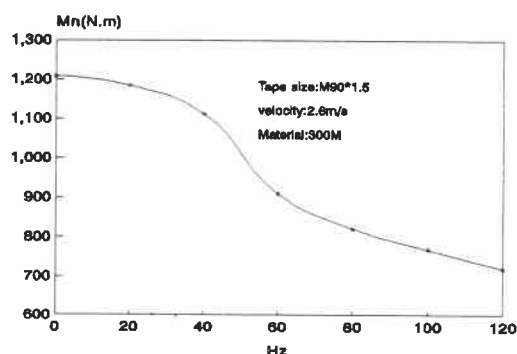


Fig.4 The variation of torque with the change of the vibrating frequency

addition, it promotes the coolant permeability which reduce the friction between the taper and the workpiece so that the tapping torque in vibrating process slows down evidently in conventional tapping. From figure 4, the vibrating machining decreases the torque nearly 50% comparing with the conventional process, the success ratio goes up to 95%.

Control system

The process of the cold form tapping is an advanced anti-fatigue technology, remarkably differing from the conventional machining. Its characteristics could be drawn as : at first, it achieves high productivity and low speed, because the threads are cold formed at one time, its productivity is up 10 to 50 times higher than cutting, especially for the long and large diameter threads, the improvement is expedient. secondly, the process is unobersvable, operator can not control the process by his experience . The third is the worse machine conditions, the cold form tapping is chipless, unusual energy is needed to overcome both the elastic and plastic deformation, meanwhile, the coolant could not penetrate into the forming zone sufficiently , so it often causes the blocking and damage of taper. Particularly, the cost of produce is rather expensive, so the effective control system is necessary to prevent rejects and decrease the cost. In this system, torque signals are gauged in process and adaptive gradient algorithm is developed.

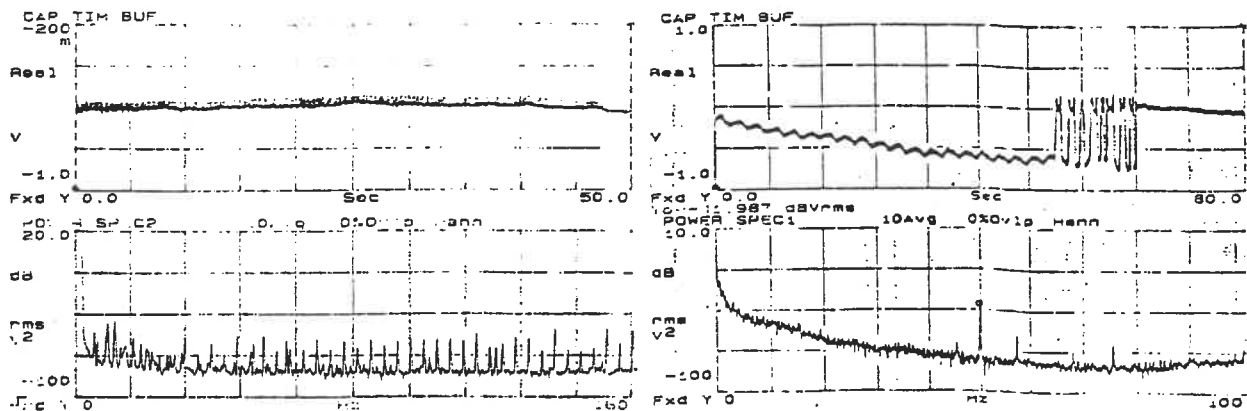


Fig.5 (a) the waveform of the torque signal in normal state (b) the waveform of the torque in abnormal state

Adaptive algorithm

The tests have been conducted on the developed machine tool and the results show that when the process is normal, the torque signals vary much stable, however, when blocking or crack of taper take place, the amplitude of the signal rises sharply shown as fig.5. By this rule, a highly effective adaptive algorithm has been developed[2].

{x} is a series of torque signals which obeys Gaussian distribution, the mean value is μ and the corresponding variance is δ . The probability density function of the signals is expressed as :

$$f(x) = \frac{1}{\sqrt{2\pi}} e^{-\frac{(x-\mu)^2}{2\delta^2}} \quad (2)$$

where the mean indicates the level of the signal's amplitude and the variance caused by the noised. A flexible recursive algorithm is exploited as follow to speed up the responding:

$$\begin{aligned}\mu_n &= \frac{1}{n} \sum_{i=1}^n x_i \\ &= \mu_{n-1} + \frac{1}{n} (x_n - \mu_{n-1})\end{aligned}\quad (3)$$

$$\begin{aligned}\delta_n^2 &= \frac{1}{n} \sum_{i=1}^n (x_i - \mu_n)^2 \\ &= \frac{1}{n-1} \sum_{i=1}^{n-1} (x_i - \mu_n)^2 + (x_n - \mu_n)^2 \\ &= \frac{1}{n-1} \sum_{i=1}^n \left[(x_i - \mu_{n-1} + \frac{1}{n} (x_n - \mu_{n-1}))^2 \right] + \left[x_n - \mu_{n-1} - \frac{1}{n} (x_n - \mu_{n-1}) \right]^2 \\ &= \frac{n-2}{n-1} \delta_{n-1}^2 + \frac{1}{n} (x_n - \mu_{n-1})^2\end{aligned}\quad (4)$$

$$\begin{aligned}\text{here } \sum_{i=1}^{n-1} (x_i - \mu_{n-1}) &= 0 \\ \begin{cases} m_n^u = \mu_n + 3\delta_n \\ m_n^l = \mu_n - 3\delta_n \end{cases}\end{aligned}\quad (5)$$

According to 3 Delta rule, when $m_n^l \leq x_{n+1} \leq m_n^u$, the process is considered as normal, on the contrary, the counter starts to work:

$$k_i = \begin{cases} k_{i-1} + 1, & \text{when } x_{n+1} < m_n^l, \text{ or } x_{n+1} > m_n^u \\ k_{i-1} + 0 \end{cases}, \quad i = 1, \dots, N \quad (6)$$

here

$$N = [\Delta T * f_s] + 1, \quad \Delta T = \frac{s_t}{v_t}$$

f_s expresses the frequency of sampling, s_t the threads pitch, v_t the velocity of workpiece. When $k_N \geq \eta * N$, $\eta \in (0, 1)$

η is determined by demand. In this paper, η is 0.85.

When abnormality take places, the microcomputer actives the control unit to rotate the spindle in reverse direction so that the taper could run out of the workpiece, the blocking and other faulty could be prevented, the taper and workpiece would be protected effectively. The figure 5 denotes the control process.

Fatigue test and Discussion

Fatigue tests are conducted on MTS-50 type material test system using the specimens of 80% height cold form taped threads and 95% height cut threads.

Test conditions: (actuating medium atmosphere)

A: loading pattern: axial sine wave

B: stress ratio: $R = \sigma_{\min} / \sigma_{\max} = 0.1$

C: frequency: 8Hz

D: temperature: 25 °C

Tables 1 shows cycle lives of cold formed thread components and cut thread component α and r denotes the remarkable degree and believable degree. x is the mean cycle lives, s the standard variance of the specimens, F the variance inspection ratio. When F is less than F_α , it means the same standard difference between the two kinds of specimens. When t is larger than t_α , it means the fatigue lives of formed thread go differently from those of cut threads. $N50$, the medium fracture life.

Table1. Cycle lives of different threads component

$\delta_{\max}$ (MPa)	Threads Type	Cycle Lives (10^3)	x	s	F	t	$\frac{[N_{50}]_F}{[N_{50}]_C}$
600	Cut	7.8, 8.7, 6.3, 5.4, 10.5	7.68	0.337	1.42 $< F_\alpha = 9.6$	33.45 $> t_\alpha = 4.60$	4.5 ~
	Taped	51.9, 36.6, 54.5, 30.1, 46.8	36.05	0.306			9.6
500	Cut	16.8, 8.5, 16.4, 14.1, 11.6	13.48	0.372	1.12 $< F_\alpha = 9.6$	46.86 $> t_\alpha = 4.86$	11.5 ~
	Taped	282.9, 279.3, 122.4, 267.3, 205.5	250.88	0.376			30.3

($\alpha = 5\%, r = 95\%$)

According to table 1, the fatigue lives of formed threads prolong 4.5 to 30 times than those of cut threads at the same stress level. Comparative analysis of the microstructure indicates that the surface integrity of former is improved greatly than that of latter. This phase of the cold form tapping is a purely elastic-plastic deforming process. As the taper comes into contact with the cored hole, its lobes traverse the surface along the spiral angle. The plastically deformed metal is displaced sideways, resulting in the surface upheaval which forms the internal threads. Along the thread form, the grains get much longer and a kind of fine fiber distribution along the teeth surface of formed thread is obtained instead of the fiber cut off. Furthermore, the plastic formed threads get residual compressive stresses in surface layer causing a significant improvement to their fatigue lives.

Conclusion

In this paper, a new anti-fatigue technology, the cold form tapping of internal threads in high strength steel, is introduced. The difference between the cold tapping formed threads and the cut threads has been investigated, and special attachment is developed. To smooth the process, vibrating machining has also been studied comprehensively, resulting in the greatly decrease of deforming torque. For the sake of expensive cost, an effective control system based on the torque signals has been established and corresponding adaptive algorithm is also discussed. At the end of this work, the fatigue tests are conducted and the mechanism on the prolonging of the fatigue lives have been discovered explicitly.

References:

- [1]. Ma Ping, The Study of the Theory and Application of In-process Monitoring Cold Form Tapping, The Master Graduation Paper, Nanjing University of Aeronautics and Astronautics, 1993.
- [2]. Ma Ping, Wang Min, In Process Monitoring Cold Form Tapping of Internal Threads in High Strength Steel, MST'94, 1994. P308-312.

DESIGN OF A LONG RANGE FAST TOOL SERVO SYSTEM USING MAGNETIC SERVO LEVITATION

Brent A. Stancil, Hector M. Gutierrez, Paul I. Ro
Precision Engineering Center, North Carolina State University
Raleigh, NC 27695-7918

1 INTRODUCTION

Magnetic Servo Levitation (MSL) has been proposed as the basis of a Long-Range Fast Tool Servo (FTS) System. Design goals call for a 1 mm range of travel, resolution on the order of nanometers, and a 300 Hz bandwidth. FTS systems using piezo-electric stacks are limited in their range of travel to millistrain elongations. Therefore, to achieve a 1 mm travel with current designs, a 1 m long piezo stack would be required. MSL can accomplish this range of motion and will introduce a new realm of possibilities in precision machining and positioning. Another advantage of MSL is that it is free of friction and backlash problems inherent in conventional mechanical actuators.

2 ELECTROMAGNET DESIGN

The electromagnets have an "E"-shaped geometry (Figure 1) with the coil wrapped around a bobbin mounted on the middle leg of the "E". The electromagnets are constructed of 0.35 mm thick laminations of an iron-silicon alloy, layered to produce a 21 mm thick stack. This alloy has been rolled such that the crystals are oriented along the major flux path direction, thus providing a higher permeability. The coil consists of 400 turns of 26 GA magnet wire. The armature is constructed of the same iron-silicon alloy laminations as the electromagnets, and is the same thickness.

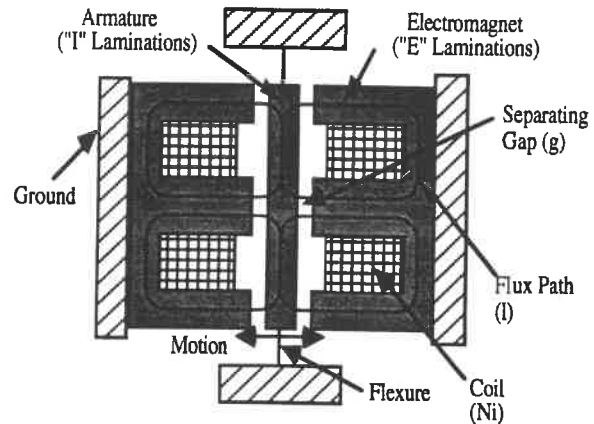


Figure 1: MSL Schematic

The design equations were derived from basic electromagnetics. The most important equation for design purposes is magnetic force. Magnetic force can be written as a function of the separating gap squared in the form:

$$F = 2 \left(\frac{Ni\mu_r}{\ell + 2g\mu_r} \right)^2 \mu_o A \quad (1)$$

where F is the magnetic force, N is the number of turns, μ_r is the permeability of the core material, ℓ is the flux path length, g is the length of the separating gap, μ_o is the permeability of air, and A is the pole face area. However, the maximum force is limited by saturation of the magnetic material. The maximum flux density (B) can be written as:

$$B = \frac{Ni}{\frac{\ell}{\mu_r\mu_o} + \frac{2g}{\mu_o}} \quad (2)$$

The iron-silicon alloy saturates at a value of $B = 2$ Tesla.

The force generated by the magnets must satisfy the inertial requirements and overcome any opposing spring force exerted by the flexures. Two pairs of magnets were chosen to double the force generated. As can be seen from Equation (1), the same result can be obtained by doubling the area. There are two reasons for choosing two smaller magnets over one large one. The first is a heat transfer issue. Heating is a definite problem due to the amount of wire and the currents involved. By making two separate magnets, there will be more surface area for convection. Secondly, by splitting the magnets, there will now be a path for a laser beam to provide for position feedback to the controller. This aspect will be described further in the next section.

3 MECHANICAL DESIGN

In any precision motion device, high stiffness is a necessity for success. To achieve the desired 0.5 mm amplitude at 300 Hz, the actuator had to be designed such that it had minimum mass for a high natural frequency of the system, while retaining structural stiffness. Part of the moving structure had to be the iron-silicon material used for the "I" laminations. The density of this material is similar to that of steel. Composites were investigated for manufacturing the rest of the moving structure, but were ruled out due to difficulty of machining. Aluminum was chosen for the moving portion of the structure. All stationary parts were fabricated in steel for maximum stiffness.

Another design difficulty associated with MSL is the fact that electromagnets only exert

an attractive force. This produces two problems. The first is the design of the moving structure. Figure 2 depicts schematically the design concept. As can be seen, there is a major portion of the structure that has no other purpose than to transmit the motion of the "I" laminations to the tool. The magnets located between the "I's" and the tool are necessary to provide the tool motion. In that sense, the gussets are necessary, yet it would be more desirable to locate the tool and "I's" in the same plane.

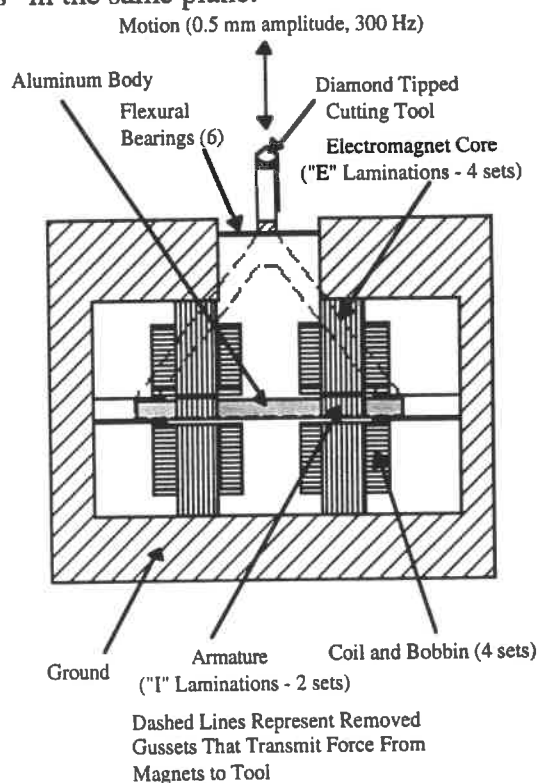


Figure 2: Actuator Schematic

The second problem is that the attractive force of the electromagnets decrease as a function of the separating gap squared (Equation 1). As a result, large magnets would be required to produce the desired force to operate the actuator. It is desirable to keep the actuator as small as possible to keep it competitive with the piezo-based FTS systems. Therefore, flexures are used to provide a

restoring force to aid in the motion. At the same time, these flexures add compliance to the system. Now, a paradox has been created. The flexures should be stiff enough that the system natural frequency is above the desired bandwidth, but compliant enough so that the electromagnets can pull against the flexures to move the armature. The design goal is a natural frequency just over 300 Hz. The stiffness required to obtain a 300 Hz natural frequency is in the same magnitude as the maximum stiffness the electromagnets can overcome to provide the desired motion. This design allows for a tradeoff of bandwidth for range if necessary.

There is a final note on the flexure design. In experiments conducted on a test fixture that only had a single pair of flexures, the second and third dynamic mode shapes were found to be within 100 Hz of the first mode. These shapes added complications to the system that could not be modeled in simulation, and were not desirable. The solution to this problem was to add another set of flexures. This moved the second and third modes to well over 1000 Hz above the first mode. This was modeled by finite element calculations and verified experimentally.

Position feedback control will be implemented in the system. A mirror will be mounted directly behind the tool with a laser mounted on the DTM separate from the FTS. This will provide the exact location of the tool to allow nanometer resolution.

4 MATHEMATICAL MODEL

Consider the magnetically-levitated system depicted in Figure 1. A mathematical model

can be formulated in such a way that nonlinearities are separable and additive:

$$a_1 \frac{d^2 x}{dt^2} + a_2 \frac{dx}{dt} + a_3 x = F_{mag2} - F_{mag1} \quad (3)$$

where x is displacement, a_i are unknown parameters, and each magnetic force F_{mag} can be described as some polynomial of coil current (i) and the corresponding air gap. For instance:

$$F_{mag2} = \sum_{j=1}^n \left(K_{1j} \frac{i_2}{(x_0 - x)^{0.5j}} + K_{2j} \frac{i_2^2}{(x_0 - x)^{0.5j}} \right) \quad (4)$$

In these equations, x_0 is the nominal air gap, and K_{ij} are parameters to be determined. The problem to be addressed here is to find the unknown coefficients involved in (3) and (4) such that experimental input-output sequences can be fitted by (3) in a minimum mean-square error sense. To use linear parametric techniques to find optimal fits for these parameters, each rational term in (4) can be considered a separate input to the system, in such a way that (3) can be written as:

$$\frac{d}{dt} \begin{bmatrix} x \\ \frac{dx}{dt} \end{bmatrix} = \begin{bmatrix} 0 & I \\ -\frac{a_3}{a_1} & -\frac{a_2}{a_1} \end{bmatrix} \begin{bmatrix} x \\ \frac{dx}{dt} \end{bmatrix} + \begin{bmatrix} 0 & 0 & \dots & 0 \\ K_{11} & K_{21} & \dots & K_{4n} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ \dots \\ u_{4n} \end{bmatrix} \quad (5)$$

The parametric methods to be considered are called prediction error methods and are based on the minimization of a cost function which is the sum of the squared prediction errors. The coefficients of such models can be grouped in a vector of unknown parameters (q). The Gauss-Newton algorithm for updating q is defined as a function of both the Hessian (H) and the gradient (g) of the error cost function with respect to q : $\theta_{i+1} = \theta_i - sH^{-1}g(\theta_i)$. Parameters of the magnetically-levitated servo system (5) were estimated using the Gauss-Newton algorithm. Since it was not possible to find

a given set of parameters that would fit all different sets of data under different experimental conditions, a recursive adjustment of the model parameters was found necessary. A recursive identification algorithm that avoids matrix inversions is:

$$\begin{aligned}\hat{\theta}(t) &= \hat{\theta}(t-1) + K(t)[y(t) - \hat{y}(t)] \\ \hat{y}(t) &= \phi^T(t)\hat{\theta}(t-1)\end{aligned}\quad (6)$$

where $y(t)$ is the observed output at time t and $\hat{y}(t)$ is a prediction of $y(t)$ based on observations up to time $t-1$. A simple approach to compute $K(t)$ is to assume that the "true" parameters change as a white gaussian process. A natural choice would then be the use of a Kalman algorithm to find $K(t)$. This approach can be modified if the estimation of the underlying covariances is difficult, by discounting old measurements exponentially in such way that an observation that is t samples old carries a weight that is l^t of the weight of the most recent observation. In this context, $t = 1/(1-l)$ can be conceived as the memory horizon of the approach. This yields:

$$\begin{aligned}K(t) &= \frac{V(t-1)\phi(t)}{\lambda + \phi(t)^T V(t-1)\phi(t)} \\ V(t) &= \frac{1}{\lambda} \left(V(t-1) - \frac{V(t-1)\phi(t)\phi(t)^T V(t-1)}{\lambda + \phi(t)^T V(t-1)\phi(t)} \right)\end{aligned}\quad (7)$$

Equations (7) constitute the Forgetting Factor approach with forgetting factor l . This algorithm was implemented to fit experimental input-output data from the magnetically-levitated system, using $l=0.98$. The model was initialized by using a vector of parameter estimates and its covariance matrix $V(t)$ generated "off-line" by the Gauss-Newton algorithm. Results are shown in Figure 3.

It can be noticed how measured response and model predictions virtually overlap. Autocorrelation of the error stays within a 95% confidence interval to that of a white process, and crosscorrelating the output with each input showed independence to within 95% confidence.

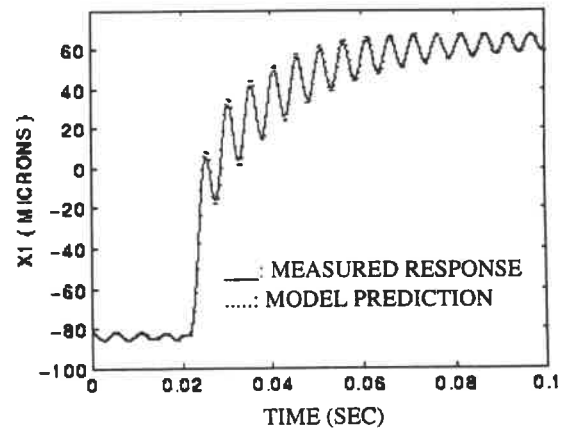


Figure 3: Step response

Recursive parametric system identification successfully models the dynamics of the magnetically-levitated fast tool servo over a wide range of displacements and frequencies. Prior knowledge of the system can be incorporated, both eliminating some fundamental system identification problems such as determination of model order and model structure and by using physical characteristics of the system to facilitate convergence and help avoid local minima. The model is linear and exact, and the calculations involved are fairly simple and can be performed within the control loop without major sacrifice in closed-loop bandwidth. Since the resulting structure is linear time-varying, several linear controller design techniques are readily available to use with this model.

A High Precision Five-Degree-of-freedom Micropositioner Based on Electromagnetic Driving Principle

WanJun Wang and Tian He

Department of Mechanical Engineering, Louisiana State University, LA 70803

Ilene Busch-Vishniac

Department of Mechanical Engineering, The University of Texas at Austin, TX 78712

I. Introduction

High precision, micropositioning devices are very essential in such applications as high precision microprobing, probing and analysis of integrated-circuit structures, manipulations of individual cells and microorganisms, microsurgery, and testing of micromachines. Since mechanical friction is one of the most important factors limiting the resolution and accuracy of high precision positioning devices, magnetic levitation technology has been used by some researchers to eliminate friction and achieve high resolution [1]. The main disadvantage of the magnetically levitated, free-floating positioning devices is that they are inherently unstable without closed-loop control. Their accuracy and resolution are therefore limited by the sensing systems and require heavy computation for digital control.

We present a five-degree-of-freedom high precision micropositioner based on spring suspension and electromagnetic driving. Since the system is inherently stable and there is no mechanical friction, the open-loop resolutions of the device are found only limited by the resolution of the 12-bit D/A board used.

II. Design and Analysis of the Micropositioner

The schematic design of the micropositioner is shown in the diagram of Fig. 1. The micropositioner consists of two parts: a moving part and a stationary part. The moving part, named as "motor", is formed with a rigid frame and three groups of coils fixed on it. The stationary part of the device, called "stator", includes a chassis and twelve U-shaped magnetic "shoes" fixed on it. Each "magnetic shoe" consists of two rare-earth permanent magnets facing each other in opposite poles fixed on a steel bracket used as magnetic flux circuit. The "motor" is attached to the "stator" with flat springs whose linear suspension allows the manipulator to move in all dimensions except the rotation along z axis. The coils have been laid out in such a way that fractions of them pass through the air gaps between the corresponding facing magnets. When electrical currents are supplied to the coils, the resulting Lorenz forces drive the "motor" to move in the five-degree-of-freedom allowed by the spring suspension. The cross-talk between movements in different dimensions and the nonlinearity are minimized by careful design of the geometry of the coils and the magnetic "shoes".

To simplify the modeling of the system, the following assumptions were made: the magnetic fields in air gaps are uniform; the "motor" itself is a rigid body and is symmetric with respect to the body coordinate system x-y-z; there is no rotation within the x-y plane; and finally the fringe effects and damping are negligible. With these assumptions the following equations can be obtained for the system:

$$\ddot{X}_0 + \frac{K_x}{M} X_0 = \frac{4BNL}{M} i_1 + \frac{BNL}{M} \sin \psi (i_4 + i_6) \quad (1)$$

$$\ddot{Y}_0 + \frac{K_y}{M} Y_0 = \frac{4BNL}{M} i_2 - \frac{BNL}{M} \sin \phi (i_3 + i_5) \quad (2)$$

$$\ddot{Z}_0 + \frac{K_z}{M} Z_0 = \frac{BNL}{M} (i_3 + i_4 + i_5 + i_6) \quad (3)$$

$$\ddot{\phi} + \frac{K_\phi}{I_{xx}} \phi = \frac{BNLd}{I_{xx}} (i_4 - i_6) \quad (4)$$

$$\ddot{\psi} + \frac{K_\psi}{I_{yy}} \psi = \frac{BNLd}{I_{yy}} (i_5 - i_3) \quad (5)$$

where X_0, Y_0, Z_0 are the displacements of the center of mass of the micropositioner in the X, Y, Z dimensions respectively; ϕ , and ψ are rotations around the X and Y axes; B is the flux density in

the air gap of the magnetic shoes; M is the mass of the manipulator; k_x , k_y , k_z , k_ϕ , and k_ψ are the spring constants in the X , Y , Z , ϕ , and ψ dimensions; i_1 , i_2 , i_3 , i_4 , i_5 , and i_6 are the electrical currents supplied to the coils 1, 2, 3, 4, 5, and 6 respectively (See Fig. 1); N is the number of turns in each coil; L is the length of each coil contained in the air gap of the magnetic shoes; d is the distance from the geometric center of the rotational arm to the straight portion of the coils; and I_{xx} and I_{yy} are the moments of inertia with respect to body coordinate system x - y .

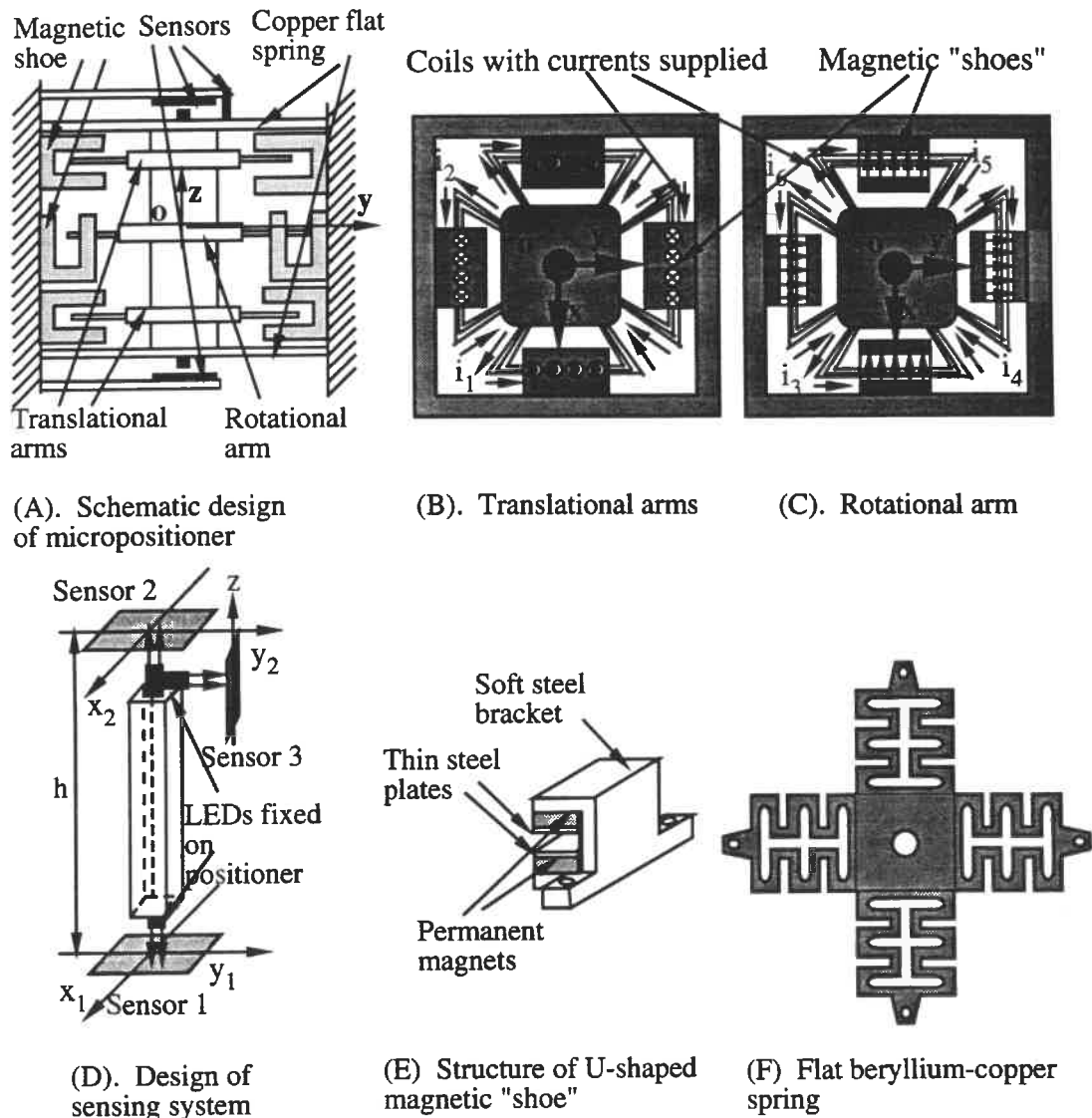


FIG. 1 Schematic Designs of Flat Springs, Magnetic "Shoes"

A sensing system using the lateral-effect position sensitive detectors (PSDs) as the sensing cells is adopted [2,3]. As shown in Fig. 1.A and D, three pin-cushion type PSDs and three LEDs are used. The LEDs are fixed on the motor and move with it, and the PSDs are fixed on the chassis. Then the movement of the motor is converted into the relative movements between the PSDs and LEDs. The light beams from the LEDs falling on the surfaces of the PSDs generates photocurrents containing the position information of the motor, which are then extracted by the signal processing circuits after adding, subtracting, normalizing, and filtering.

III. Experimental results

III.1. Open-loop Results

The open-loop calibration is carried out to determine the linearity and resolution of the micropositioning system.

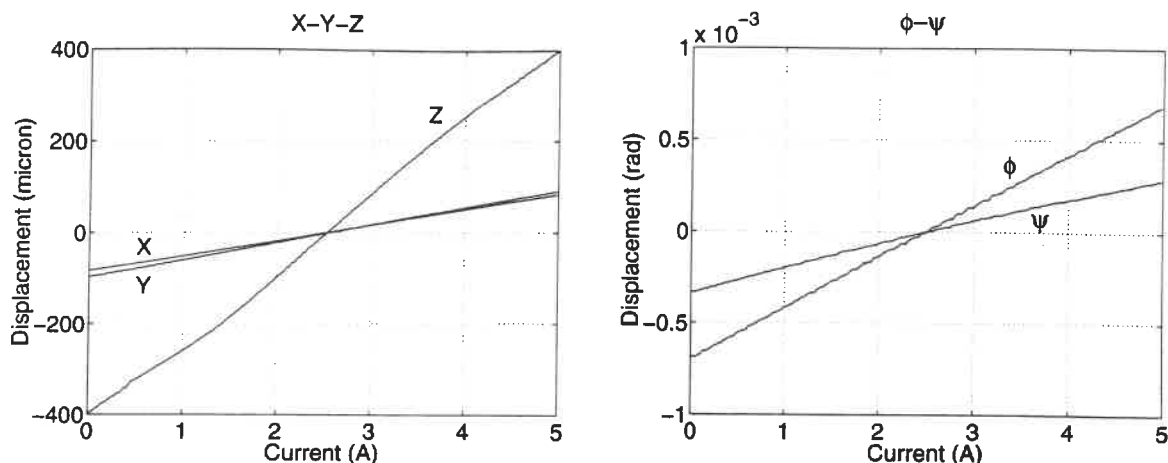


FIG. 2. The Open-loop calibration results

Fig. 2 shows the open-loop calibration results for all five degree-of-freedom (x , y , z , ϕ , ψ). The results show that the maximum working range in the x direction is $180\text{ }\mu\text{m}$, in the y direction is $180\text{ }\mu\text{m}$, in the z direction is $680\text{ }\mu\text{m}$, in the ϕ direction is $1.38 \times 10^{-3}\text{ rad}$ and in the ψ direction is $0.68 \times 10^{-3}\text{ rad}$. The system shows very high linearity. There is also very little cross talk between different dimensions, which makes it possible to control each degree-of-freedom separately.

Since the system has open-loop stability, the open-loop resolution is limited by that of the driving system. The driving circuit is found to have much higher accuracy than that of the 12-bit D/A board. Therefore the open-loop resolution is found to be about $0.05\text{ }\mu\text{m}$ ($180\text{ }\mu\text{m}/4095$) in x and y dimensions and $0.15\text{ }\mu\text{m}$ ($600\text{ }\mu\text{m}/4095$) for the z -dimension, as set by the 12-bit (0~4095) digital to analog conversion board.

III.2. Closed-loop control and experimental results

A. Step responses without feedback control

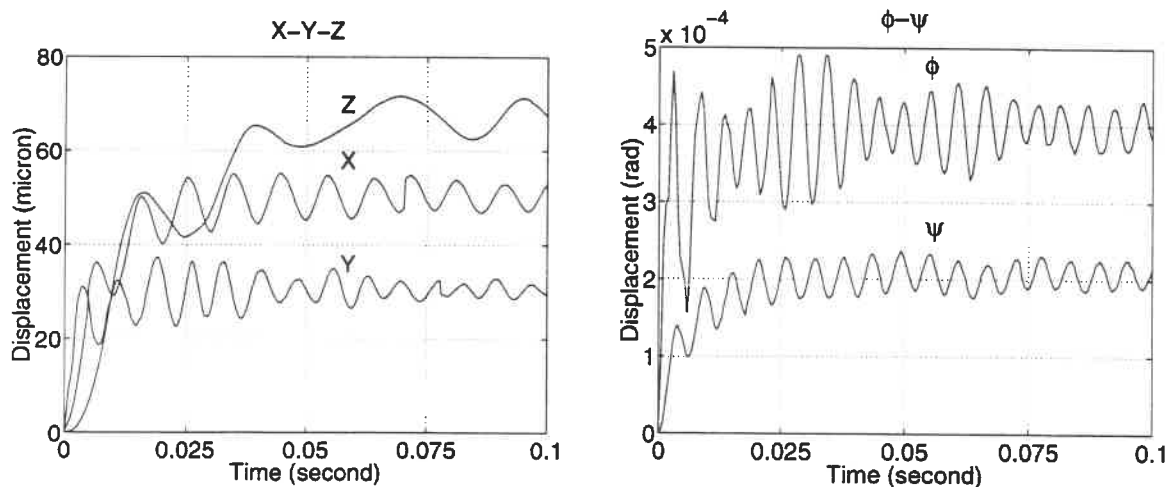


FIG. 3. Step responses of the system without closed-loop feedback control

Fig. 3 shows the step responses of the micropositioning system in all five degree-of-freedom without feedback control. It can be seen clearly from these responses that the micropositioner is basically a second-order, under-damped system. It has very high damped natural frequency, which attributes to the small mass of the manipulator and large stiffness of the springs. It also has very low damping effect.

B. Step responses with a PID controller

To overcome this undesired oscillatory response, A PID controller is designed. The step responses of the micropositioner with a PID controller are shown in Fig. 4. It can be seen that the high oscillation in the open-loop step responses has been eliminated. There are no overshoots, which ensures that the specimen or the probe will not be damaged by the collision between them during positioning.

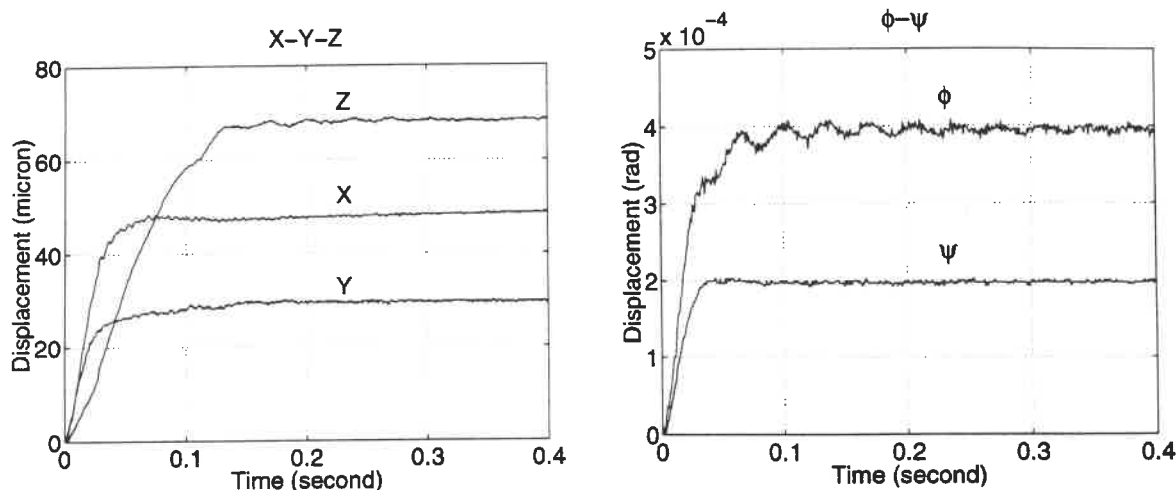


FIG. 4. Step responses of the system with closed-loop control

The sensing system used in the experiment has been calibrated with a precision stages and a translational resolution of $0.1 \mu\text{m}$ was confirmed over the working space of 1.2mm by 1.2mm . However, the 12-bit A/D conversion board used to digitize the sensing signals limits the measurement resolution to $0.3 \mu\text{m}$. This set the limitation for the resolutions for closed-loop control.

IV. DISCUSSION

Since the open-loop accuracy of the device is only limited by the resolution of the 12-bit D/A board, higher resolution can be expected if 14-bit or 16-bit D/A board is used. Larger working range is possible if higher driving current, smaller distance between the facing magnets or more turns of the coils are adopted. In conclusion, the prototype micropositioner has the advantages of high accuracy, large working space, open-loop stability, high linearity, compact size, and high load capacity. These advantages make it potentially useful in such applications as high precision microprobing, probing and analysis of integrated-circuit structures, in the manipulations of individual cells and microorganisms, microsurgery, and testing of micromachines.

REFERENCES

1. W. Wang and I. J. Busch-Vishniac, ASME WAM, 90-WA/DSC-23, Dallas, Texas, November 25-30, 1990.
2. D. Qian, W. Wang, and I. J. Busch-Vishniac, *IEEE Trans. Instrumentation and Measurements*, PP. 14-18, Volume 42 (1), February, 1993.
3. C. Narayanan, A. Bruce Buckman, I. Busch-Vishniac, and W. Wang, *IEEE Trans. Electron Devices*, Vol. 40, No. 9, September, 1993. pp. 1688-1694.

INFLUENCE OF FORM DEVIATIONS ON THE DYNAMIC PERFORMANCE OF MICROHYDRAULIC ACTUATORS

K. Sadashivappa
Research Scholar

M. Singaperumal
Associate Professor

K. Narayanasamy
Associate Professor

Department of Mechanical Engineering
Indian Institute of Technology, Madras-36, India.

INTRODUCTION

Hydraulic actuators have found wide applications due to their versatility, manoeuvrability and compactness. Miniaturised high performance piston type actuators, particularly axial piston motors have caused the development of sophisticated micro-mechanisms which find applications in many mechatronic systems [1]. In hydraulic actuators, the usage of seals on the piston surface causes positional inaccuracies owing to coulomb friction. Hence sealless pistons with sloping surfaces have been suggested. Some authors have carried out theoretical and experimental investigations, mainly related to static and dynamic friction characteristics, on these piston type of actuators [2-4].

In the manufacture of pistons, form errors like out-of-roundness, taper and surface irregularities are bound to occur. In all the above studies the effect of out-of-roundness on the pistons on the dynamic performance of the actuator has not been considered. In an earlier investigation, the authors have shown that the pistons with defined form improve slow speed characteristics of the actuators [5]. In this paper, the results of an experimental study on the effect of taper and out-of-roundness on the dynamic performance of a micro-hydraulic actuator are reported.

EXPERIMENTAL WORK

Specifications of the test motor, a swash plate type fixed displacement axial piston hydraulic unit, are given in table 1. Pistons with 3-lobe profile and taper are fabricated and introduced into this motor. Speed-time characteristics are obtained by conducting dynamic tests over a wide range of operating conditions, in a bench test rig. **Bench test rig:** A schematic diagram of the bench test rig is shown in Fig. 1. The hydraulic motor shaft drives inertia discs through a cradled in line transmission type torque transducer of capacity 50 Nm and sensitivity class A, provided with an optical speed sensor. Two pressure transducers of capacity 200 bar and 10 bar are used to measure pressure line and return line pressures respectively. Pressurised oil is supplied to the motor from a hydraulic power pack.

Fabrication of pistons: The 3-lobe profiles are obtained on the pistons by maintaining the workpiece center below the center line, in a centerless grinding operation. The intended taper is obtained by maintaining the angle of control wheel. The 3-lobe profile and taper are obtained in a single pass. The dimensions, taper and out-of-

roundness of the piston are measured using high precision micrometer and form tester. The dimensional and form details are listed in table 2. The details of existing cylindrical pistons are indicated in table 3.

The material of the 3-lobe piston is medium carbon steel. They are sur-sulf treated to improve the hardness to about 50 Rc. A profile of the 3-lobe piston is shown in Fig. 2. From the figure it is clear that there is not much deformation of the profile though the out-of-roundness has reduced from $10.4\text{ }\mu\text{m}$ to $9.3\text{ }\mu\text{m}$, whereas the hardness has increased from 29 Rc to 54 Rc and surface roughness Ra has increased from $0.12\text{ }\mu\text{m}$ to $0.38\text{ }\mu\text{m}$.

RESULTS AND DISCUSSION

Dynamic response test is conducted with different inertia loads. Load is mounted on the motor shaft and oil is supplied to the motor. The motor starts to rotate and transducer outputs are acquired simultaneously. Fig. 3 indicates the results of a typical test carried out at a supply pressure of 80 bar with 20 N inertia load. It is clear from the figure that the motor with the cylindrical piston assembly as well as the 3-lobe piston assembly reaches the maximum speed almost at the same time during run-up phase. During rundown phase, motor with the 3-lobe piston assembly takes more time than the cylindrical piston assembly to come to stand still. This indicates that the frictional resistance is less in the case of the 3-lobe piston assembly.

CONCLUSIONS

Experimental investigations on a commercial fixed displacement swash plate type axial piston motor were carried in a bench test rig by introducing specially fabricated 3-lobe pistons. The results are compared with that of cylindrical pistons. The study indicated that the dynamic response of the motor is not affected due to the inevitable form deviations on the pistons. The frictional resistance in the motor with the 3-lobe pistons is comparatively small hence the motor can be made to run smoothly at slow speeds. Thus the slow speed characteristics of the motor can be improved by having defined form on the pistons.

REFERENCES

1. K.Sadashivappa, M.Singaperumal, K.Narayanasamy, Some performance aspects of micro-hydraulic actuators, Proc. of IFTOMM Int. Micro-mechanism Symp., Tokyo, 1993, 161-171.
2. N.Meikandan, R.Raman, M.Singaperumal, K.N.Seetharamu, Theoretical analysis of tapered pistons in high speed hydraulic actuators, Wear, 1990, 137, 299-321.
3. Akira Hibi, Starting torque of swash plate type axial piston motor, Bull. of JSME, 1974, Vol. 17, No. 106, 479-486.
4. N.Meikandan, R.Raman, M.Singaperumal, Experimental study of friction in hydraulic actuators with sealless pistons, Wear, 1994, 176, 131-135.
5. K. Sadashivappa, M. Singaperumal, K. Narayanasamy, On the efficiency of axial piston motor considering piston form deviations, communicated to J. of Mechatronics, UK, for publication.

TABLE 1 : HYDRAULIC MOTOR SPECIFICATIONS			
1	Geometric displacement	ml/rev	1.9
2	Swash plate angle	deg.	20
3	Number of pistons		9
4	Diameter of piston	mm	6
5	Piston stroke	mm	7.8
6	Pitch circle diameter	mm	21.3
7	Piston length	mm	24

TABLE 2 : 3-LOBE PISTON DETAILS					
Bore	Piston	Type	Roundness μm	Taper μm	Radial Clearance μm
1	6	D	09.3	8.8	16.9
2	12	D	13.8	6.5	15.9
3	28	D	12.1	8.8	15.1
4	8	D	12.8	6.4	18.8
5	26	D	13.2	6.8	16.9
6	2	D	15.1	8.2	16.2
7	24	D	10.3	2.2	19.9
8	9	D	13.5	8.0	16.6
9	18	D	10.1	3.9	12.9

TABLE 3 : CYLINDRICAL PISTON DETAILS					
Bore	Piston	Type	Roundness μm	Taper μm	Radial Clearance μm
1	1	C	1.2	0.7	14.6
2	2	C	1.0	1.2	18.8
3	3	C	1.2	0.7	18.0
4	4	C	2.5	1.0	18.2
5	5	D	1.1	0.4	18.6
6	6	C	0.9	1.4	20.4
7	7	C	1.5	0.3	19.5
8	8	D	1.0	0.2	23.5
9	9	C	1.0	0.9	13.5

Note: C = Converging clearance, D = Diverging clearance

Pr. Tr. : Pressure Transducer
 STI : Speed Torque Indicator
 HM : Hydraulic Motor

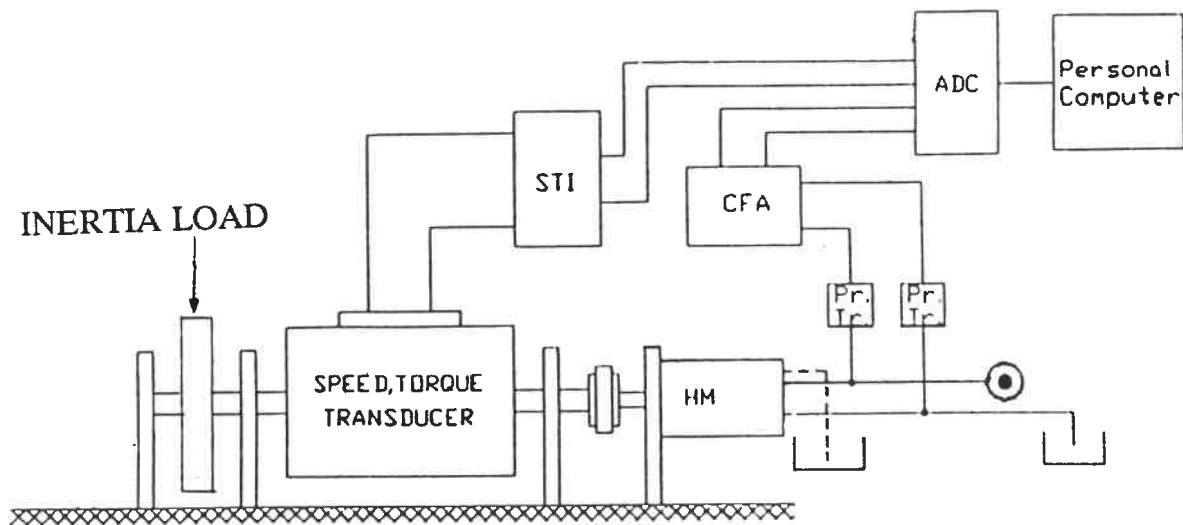


Fig. 1 : Schematic diagram of the test rig and data acquisition system.

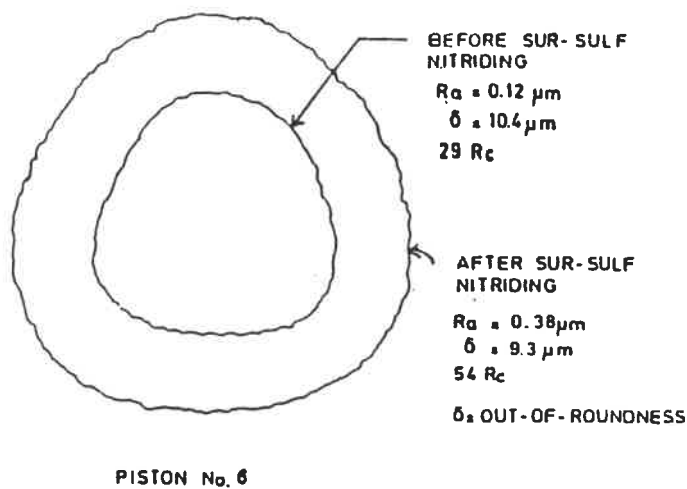


Fig. 2 : 3-lobe piston profile.

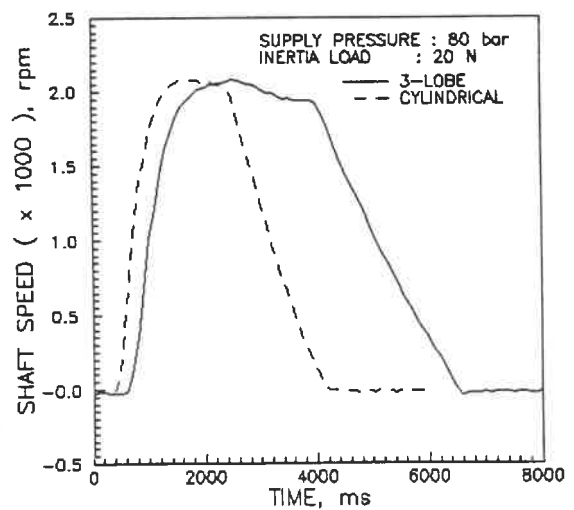


Fig. 3 : Dynamic response.

Development of a Parallel Architecture Five-Axis Coordinate Measuring Machine

Bernhard Jokiel, Jr.
John C. Ziegert

Machine Tool Research Center
University of Florida
Gainesville, Florida 32601

Introduction

Coordinate measuring machines (CMM) are widely used in industry for dimensional inspection of manufactured parts. Typical CMMs are designed using a kinematic structure consisting of a serial chain of three mutually perpendicular prismatic joints to achieve three degrees of freedom in motion. The actuated structure is used to guide a probe around the contours of a work piece held securely on a table below. The position of the probe (touch trigger, capacitance or optical) is determined by the position sensing system (typically glass scales) built into the kinematic structure itself. This widely used and accepted architecture has many short comings which limit its ability to reach its achievable accuracy without resorting to complex and expensive techniques to remove errors induced by thermal and dynamic effects, and component geometry. An alternative approach is to use a parallel kinematic structure similar to the well known Stewart platform. This paper describes a conceptual design which will allow CMM's to be made less expensive, more accurate, and with a greater robustness to thermal and elastic deformations.

Design requirements for precision machines

Design of a machine to perform precision dimensional measurement requires several basic tasks to be completed. First, one must establish a length reference or metric and a means of transferring or establishing that metric in the work space of the machine. Second, one must establish a reference coordinate system whose origin and geometry are known. Third, one must be able to generate repeatable motions of the probe relative to the work piece in a manner such that these motions can be measured relative to the reference coordinate system using the metric. Fourth, the characteristics of the probe must be known, since it links the measuring machine to the part being measured. The accuracy of the machine will depend on the degree of success in accomplishing each of these tasks.

Teague[1] has identified a number of design principles which will aid in accomplishing these tasks in an optimal manner. These principles include isolation of the instrument from disturbing environmental effects, kinematic mounting of mating assemblies, satisfaction of the Abbe' alignment principal, and use of metrology frames.

Design requirements for serial kinematic structures

A precision machine with a serial kinematic structure (i.e. a conventional Cartesian machine) has the following design requirements:

1. The guideways should be as straight as possible. This is necessary since it is

impossible to satisfy the Abbe' alignment principle for all three axes. Therefore, extremely straight guideways are needed to prevent the moving bodies from rotating and causing displacement errors.

2. The machine elements should be very stiff. Creation of very straight guideways does not help if they sag under the weight of the moving bodies which they must support. This leads inevitably to large, heavy structures; qualities which are detrimental to the dynamic performance of the machine.

3. The guideways must be aligned very precisely. The guideways essentially form the reference coordinate system for displacement measurements in machines without a separate metrology frame. Therefore, if they are not arranged to be perfectly orthogonal to each other, measurement errors will occur.

4. A reliable and accurate displacement measuring system is required for measuring displacements of the individual slides along the guideways.

Design requirements for parallel kinematic structures

It is interesting to compare the design requirements for accuracy in parallel kinematic structures of the Stewart Platform type with those for serial kinematic structures. In parallel structures, the position and orientation of the moving platform are obtained from the solution of a set of geometric relationships. The inputs to these relationships are the geometry of the base and moving platform (i.e. the coordinates of the centers of the spherical joints), and the absolute distances between joint centers. Therefore, in order to achieve accuracy in a parallel mechanism one needs to realize the following requirements:

1. The spherical joints should produce perfect spherical motion. The actuators must rotate about fixed points on the base and platform.

2. The absolute distance between the centers of corresponding spherical joints on the base and platform must be measured with a high degree of accuracy. In general, displacement measuring devices will be used to measure the changes in length of the individual legs. Therefore, a system for determining the initial lengths, must be developed.

3. The geometry of the base and platform must be stable and known to high accuracy. Since the position of the probe is dependent on the coordinates of the joint centers on the base and platform, it follows that any deformations of these bodies will cause errors in that position. Therefore, the base and platform should be rigid and thermally stable. This requirement can be relaxed if it is possible to monitor the actual positions of the joint centers while the instrument is in use.

It is clear that the first two requirements are descriptions of the functional capabilities of the laser ball bar (LBB) which is essentially an extensible prismatic strut with spherical joints on the ends. Using laser interferometry, the LBB is able to measure the distance between the centers of the spherical joints. The trilateration procedure utilized to measure the spatial coordinates with an LBB can be viewed as a degenerate form of Stewart platform mechanism. If the moving platform of such a mechanism shrinks to a single point so that all of the joint centers on it become coincident a tetrahedron is formed. The LBB sequentially measures the lengths of the sides of this tetrahedron to obtain the spatial coordinates of the apex. Thus, the LBB naturally satisfies two of the design requirements for accurate parallel mechanisms, and can be viewed as a building block towards the creation of platform type mechanisms with high positioning accuracy.

Conceptual hexahedral design

The proposed structure (Figure 1) consists of a triangular granite base with columns rising from each of the three corners. A circular ring is kinematically supported by the three columns. Three precision spheres are affixed on the interior of the ring, and form the reference coordinate system. Two LBBs emanate from magnetic sockets on each of the three base spheres to form the legs of two tetrahedrons, one above and one below the plane of the ring. The spheres at the apexes of the two tetrahedrons are connected by a rigid center rod. A

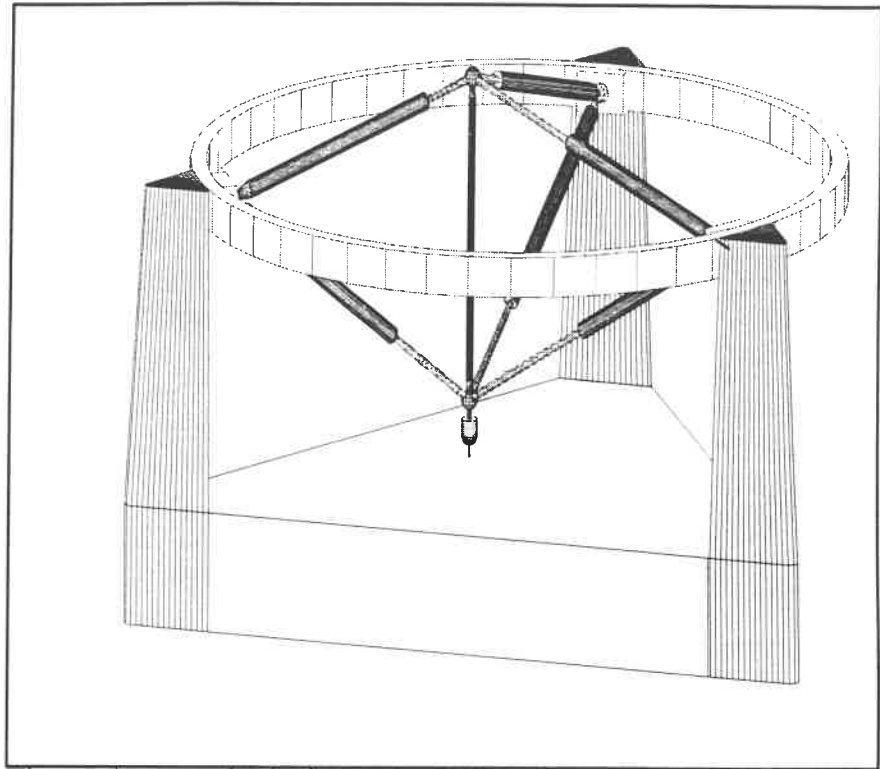


Figure 1 - Hexahedral, 5-axis CMM concept

probe (touch trigger, optical or capacitance) is attached to the lower sphere such that the probe tip is collinear with the center rod. The coordinates of centers of the upper and lower spheres are easily computed if the leg lengths and base lengths are known. The probe tip coordinates may then be computed since its center is a known distance along the line joining the upper and lower spheres. The workspace of the machine with the center rod held vertical is shown in Figure 2.

The center rod may be guided by an external robot, thus using the hexahedron as a passive metrology frame. Alternatively, the individual legs may be self-actuated via pneumatic, hydraulic, or mechanical means.

No matter which method is used, the problem of determining the base lengths and initial leg lengths remains. The position of the probe tip is a function of the three base lengths and all six leg (LBB) lengths from sphere center to sphere center. When the laser interferometers in each leg are turned on, the leg has some finite length which is unknown. The initial leg lengths and the base lengths must be determined in order to find the total distance between sphere centers which is needed to compute the coordinates of the probe.

The hexahedral CMM has only five degrees of freedom of motion, but six leg length sensors. This redundancy may be exploited to determine the initial leg and base lengths. The distance between the top and bottom sphere centers is constant, allowing one to formulate an expression for this distance in terms of the unknown initial leg and base lengths, and the measured leg length changes as the machine is exercised through a number of different positions. Using this data, one can formulate a least squares problem to find the set of base lengths and initial leg lengths which minimize the variation in the computed

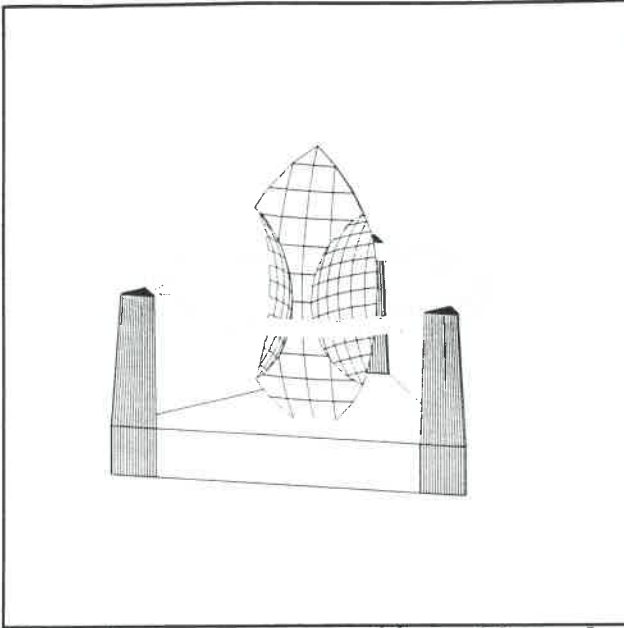


Figure 2. Locus of attainable positions of upper and lower sphere.

center rod length. A preliminary error budget for the machine was performed. The predicted spatial positioning accuracy is $\pm 2.8\mu\text{m}$.

An interesting by-product of this self-initialization method is that it is possible to actively detect and correct for thermal drifts and structural deformations during measurement cycles. During a measurement cycle, the computer looks at a moving window of LBB length data, and computes a new set of updated initial leg and base lengths. This revised set of lengths then is used to calculate the next positions of the probe. If the machine were to thermally expand during a measurement cycle, the change would be reflected in the data collected from the LBBs, and would be automatically compensated for by the machine initialization routine. This

continuous self-initialization procedure has the potential to make the accuracy of the hexahedral CMM essentially insensitive to its thermal environment, thus allowing it to be placed directly on the shop floor, obviating the need for a temperature controlled room. In addition, the hexahedral CMM can be expected to have significantly better dynamic performance since the mass of the moving elements is much lower than in conventional designs.

References:

1. Teague, E. C., "Tutorial on Precision Instrument Design", ASPE Annual Meeting, 1994.

Acknowledgements:

This material is based on work supported in part by NSF Grants # DDM-9358138 and GER-9354980.

Micro-Impulse Generation Digging Tool for Micro Surgery

Masayuki Nakao, Yotaro Hatamura, Ryosuke Asai and Takashi Yano
The Department of Engineering Synthesis, The University of Tokyo
7-3-1 Hongo Bunkyo-ku, Tokyo 113, Japan

1 Introduction

Even a skillful surgeon has much trouble doing a micro surgery. When he handles a tool by micro-force on a patient, he has to slightly move his arm, hands or fingers. But he may tremble his hands with a strong tension of the muscles [1]. Finally, he may overload a tool because the human cannot control the micro-force.

In some micro surgeries, he must carefully remove bones, lymph or skins of the patient. For examples, he removes bones of a middle ear to release a swollen facial nerve. At that time, he must control several 10 mN to remove the bone pieces of 0.1mm in diameter under microscopic observation.

To execute the micro surgery, employment of an automatic robot is not a good decision. Because the patient deeply breathes, the robot can't fix his coordinates for its motion control. Because the surgeon always changes the operation process in order to cope with the condition of the patient, the programmed robot can't work safely. Development of a smart tool handled by the surgeon's judgment should be better for future micro-surgery.

In order to assist the micro surgery, the authors propose a micro-force control tool. It is composed of a micro-force generator and a micro-force sensor in its pencil-like shape. It can also control a micro-impulse when it efficiently dig, chisel, cut and peel the patient under vibration. For example, it chisels bones under controlling the impulse using a piezo element instead of hammer hitting.

Any surgery tool must be essentially safe against infection and overload. The tool has a changeable blade of the chisel for easy disinfection. And it has a fail-safe mechanism against excess force. Its rigidity dramatically changes smaller when it works over the critical force. In this paper, we introduce the tool called as a "micro-impulse generation digging tool" along the design process.

2 Design process of the tool

We design the tool with a "Creative Design Principle (CDP)". The CDP is a design theory shown in Fig. 1(a). It has the following designer's thinking process:

deciding on requirement, analyzing for function, decomposing to functional elements, mapping to mechanism elements, developing of structure elements and synthesizing to total structure. Of course, the process direction may reverse if the designer thinks that a mechanism configuration doesn't satisfy each of the functions, or that the structural elements can't be fabricated.

The design process of the tool is shown in Fig. 1(b). As mentioned above, the requirement of the tool is to assist the micro surgery. To realize the requirement, it needs (1) controllability the micro-force and (2) safety against overload and infection.

To satisfy the functions, we analyze the following functional elements: (1) micro-force generation, (2) micro-force sensing, (3) fail-safety against overload and (4) ability of disinfection. Next we realize physical figures of their conceptual elements. At this time, an one-by-one mapping from functional element to mechanism element is important because non-redundant machines should be composed of independent and minimum functional elements [2]. We select the mechanism elements for the mentioned functional elements, respectively, as shown in Fig. 1(b).

Through the development of structural elements and the synthesis to total structure, we realize the tool as shown in Fig. 2. The generated micro-force is flowed along the two traces: from piezo actuator to changeable blade, and to force sensor, fail-safe structure, cover and surgeon's hand. Z-direction is defined as shown in the figure. The dimension is 170 mm in length and 29×18 mm in cross section. Its weight of 135 g is easy to operate.

3 Evaluation of the tool

3.1 Evaluation of the structural elements

At first, the micro-force sensor is evaluated. The piezo-type sensor (Kistler # 9131A) can detect only force at z-direction with fine resolution of 5 mN. The sensor is good at compactness ($\phi 7\text{mm} \times 3\text{mm}$) and high rigidity ($700 \text{ N}/\mu\text{m}$). But the output is reduced by about 100 mN/sec because generated elec-

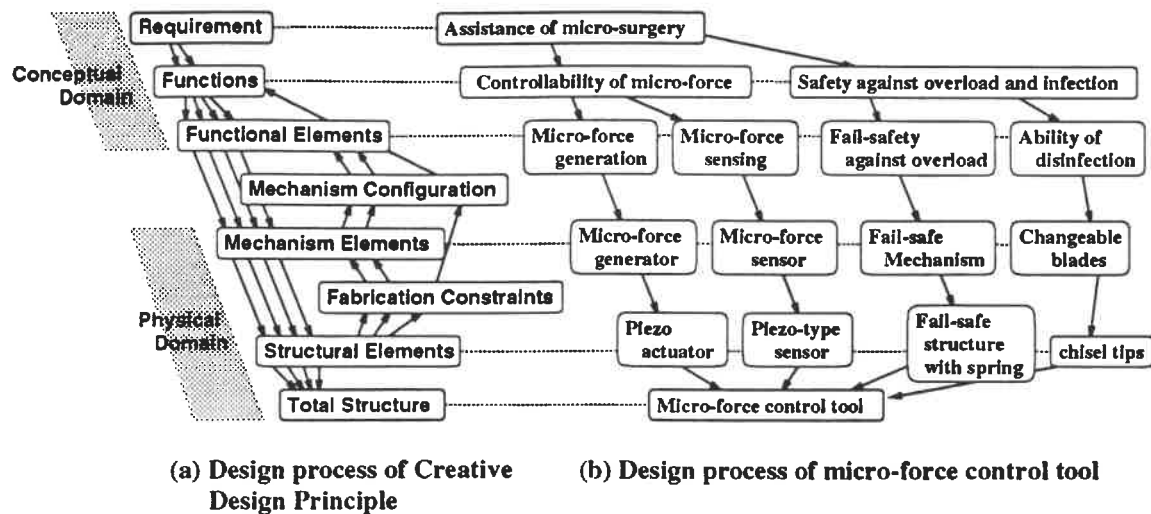


Fig. 1 Design process of micro-force control tool

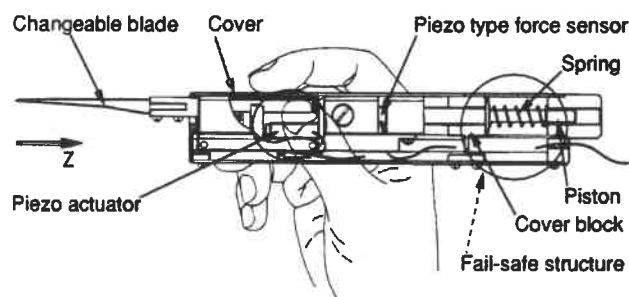


Fig. 2 Schematic of designed micro-force control tool

tric charges are disappeared. We conduct a digging experiment shown in Fig. 3 to evaluate the sensor. We measure the z-direction force on the tool (F_{zt} , represented by electric charge) measured by the piezo-type sensor and the z-direction force worked on a bone (F_{zb}) measured by a strain gauge type force sensor. From the result shown in Fig. 4, it is verified that the step force on F_{zb} changes a differential-like force on F_{zt} . Electric charge on F_{zt} is generated by a change of force.

Secondly, the micro-force generator is evaluated. The piezo element (Tokin $6 \times 6 \times 20\text{mm}$) can generate 870 N under input voltage (V_i) of 100 V. The tool has a lever structure of 5 times magnification of displacement. The example of the variation of F_{zt} , F_{zb} and V_i is shown in Fig. 5. A signal of 0-75V at 60Hz can make a force variation of 50-125mN at bone. As the F_{zt} waveform is similar to the variable part of the F_{zb} waveform, it is verified that the output reduction on F_{zt} was negligible at 60 Hz.

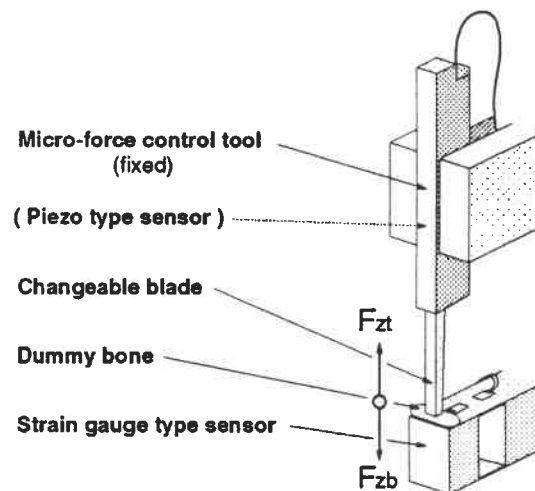
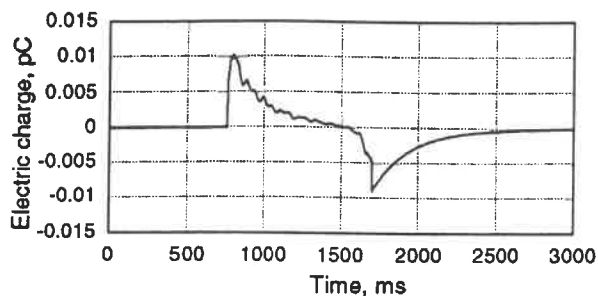


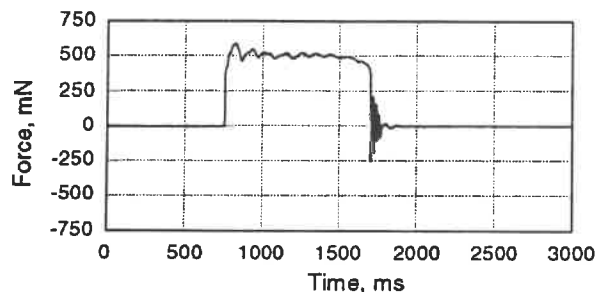
Fig. 3 Schematic of digging experiment

Under a constant frequency of V_i , we can get the quantitative relation between variable force and electric charge. The small vibration at about 500Hz on F_{zt} and F_{zb} is caused by resonant vibration of the strain gauge type sensor because of its low rigidity ($0.007\text{N}/\mu\text{m}$).

Thirdly, the fail-safe structure is evaluated. The critical force is initially fixed as 1.8 N using a compressive spring. The relation between force acting on the tool by weights and z-displacement of the blade is shown in Fig. 6. Under 1.8 N, the rigidity is large by $0.6\text{N}/\mu\text{m}$ because a piston is pushed on the cover block by the spring. But over 1.8 N, the rigidity becomes smaller to $0.0005\text{N}/\mu\text{m}$ because the piston is moved away from the cover block.

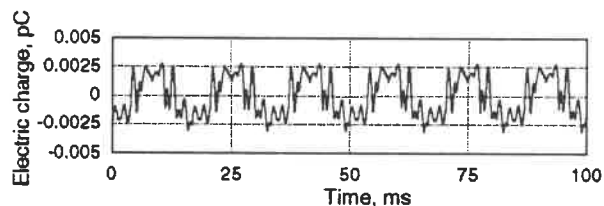


(a) Force acting on the tool = F_{zt}
(sensed by piezo type sensor)

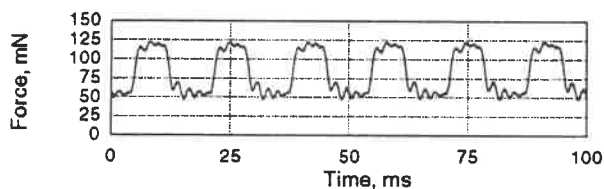


(b) Force acting on the bone = F_{zb}
(sensed by strain gauge type sensor)

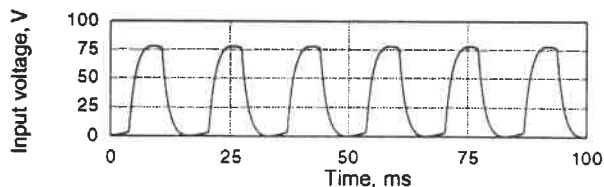
Fig. 4 Measured forces in acting a step force



(a) Force acting on the tool = F_{zt}
(sensed by piezo type sensor)



(b) Force acting on the bone = F_{zb}
(sensed by strain gauge type sensor)



(c) Input voltage for driving piezo actuator = V_i

Fig. 5 Measured force in acting continuous impulses

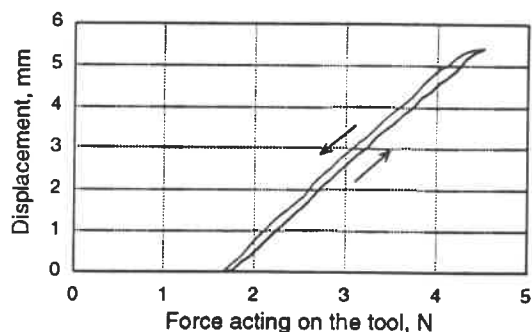


Fig. 6 Relation between forces acting on the tool and tool's displacement

3.2 Evaluation of the total structure

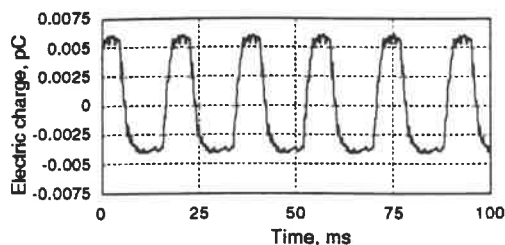
To evaluate the utility of the tool, we measured an influence induced by holding the tool in hand. The difference of force acting on the tool between fixed and hand-held under 0-75V at 60Hz without the strain gauge type sensor is shown in Fig. 7. The signal waveform of the fixed one is very similar to that of input voltage shown in Fig. 5(c). But in hand-held one, gradient of the signal waveform becomes gentler than that of the former. It can be identified that the difference is caused by the force absorption in soft finger / hand / arm of the operator.

4 Discussion

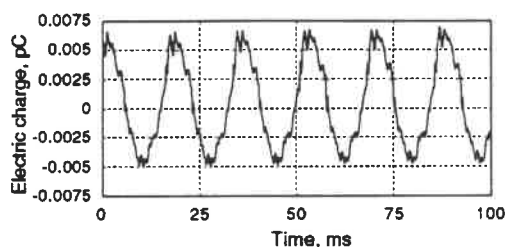
4.1 Necessity of micro-handling on short / hard force loop

When chiseling a bone with maximum impulses, the operator has to fix the tool against the counter force. At this time, it is useful for him to place his little finger and back of the hand on a fixed table. Without the table, the counter force should cause the fatigue of muscle of arm and shoulder. Micro-handling needs a shorter force loop as shown in Fig. 8(b).

Moreover, to prevent the force absorption as mentioned in Fig. 7, it is better to directly fix the tool on a rigid arm as shown in Fig. 8(c). The arm could become movable easily when the surgeon judges that the patient is dangerous. The hard force loop with a rigidity-changeable arm is promising for micro-handling.



(a) Force acting on the tool (fixed)



(b) Force acting on the tool (hand-held)

Fig. 7 Difference of forces acting on the tool between "fixed" and "hand-held"

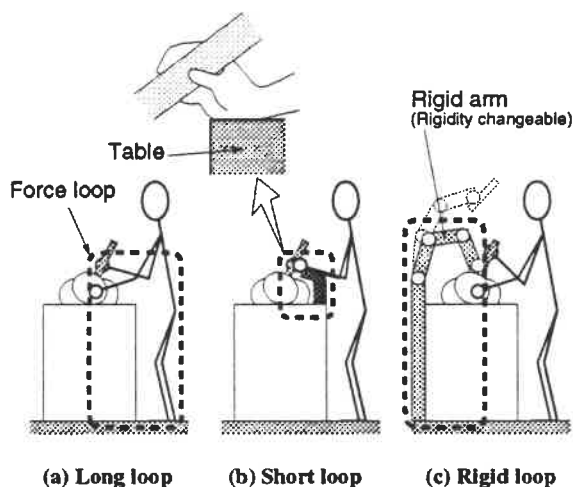


Fig. 8 Schematic of various force loops

4.2 Necessity of force control for tele-surgery

Tele-surgery is a necessary technology for humans. For examples, a tele-surgery through key-holes of the skin with endoscopic observation has been required. A tele-surgery beyond a blood fence will be also required because the surgeon worries about the patient's blood at most. Here, a transmission of force

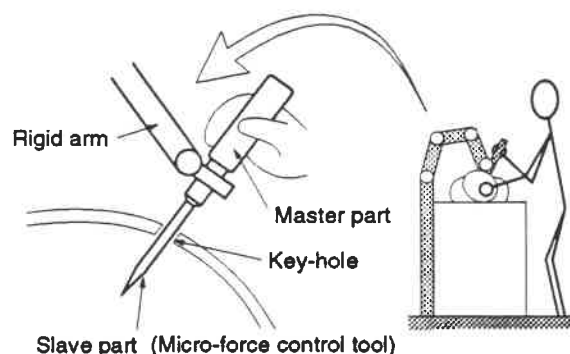


Fig. 9 A tele-surgery system with micro-force control tool

information is inevitable. We know that medical doctors touch patients in order to judge the inside diseased organ. Because the tool also can be worked as a sensor detecting rigidity, it will be indispensable for tele-surgery.

When the tool is used as a slave part as shown in Fig. 9, it can realize a bilateral force control. A configuration including a new master part and the rigid arm should be a future tele-surgery system.

5 Conclusion

To assist micro surgery, a micro-force controllable tool is designed, fabricated and evaluated. We confirmed that the tool could generate micro-impulse for digging / cutting / chiseling bones with the piezo type force sensor and the piezo actuator. It is safe against overload handling using a newly designed fail-safe mechanism. With the tool, surgeons can employ micro-force handling which is difficult for humans to control. In the future, the tool will be promising for micro surgery and tele-surgery.

We thanks Prof. Tetsuo Ishii and Prof. Mikiko Takayama of Tokyo Women's Medical College for its evaluation.

Reference

- [1] M. Nakao, et al.; Proc. of ASPE 94, 61-64
- [2] Nam P. Suh; The principle of design, Oxford University Press

DESIGN OF SPUR AND HELICAL GEARS

Josef MANDÁK - DLI MTA in Brno

ABSTRACT: The paper presents the results obtained from the deformation and stress analyses of spur and helical gears. Insufficient bending and contact strength are major causes of failure and contact damage in gear teeth and have been the reason of research on gearing for a long time. Each design of one or two gears in mesh usually begins with the characteristics of the rack and goes on with the calculation and design of the gears.

For calculating tooth stress in standard types of spur and helical gears are used the German DIN 3990, the Niemann's methodology, the AGMA standard, the Lewis' methodology, the Bach's methodology, the Dudley's methodology, the Henriot's methodology, the Buckingham's methodology, the Merrit's methodology etc., which determine the stress state in the critical cross-section of the tooth or in the area of contact two gears in mesh (they all have major advantages and disadvantages). In all cases the problem is to determine the position of the tooth's weakest (critical) section at the fillet. Some of above-mentioned design methods use approximate solutions. The approximate methods use correction factors to compensate for some of the assumptions made during the simplification of the problem since most of the correction factors (the nominal stress on the weakest tooth's section is calculated and multiplied by experimental coefficients, like the stress-concentration factor - correction factor, to simulate the real stress situation e.g.) are obtained empirically and because the analytical methods are difficult to be used in everyday life engineering practice.

For calculating tooth stress in non-standard types of spur and helical gears are used the Finite Element Method (FEM), the Boundary Element Method (BEM), Combined BEM-FEM method (Brady and Wassynig), Global Local Finite Element Method (GLFEM), Elasticity method and conformal mapping etc., with respect to the basic generally valid elastic relations among basic quantities - displacements, relative elongations, strength and load states.

Design of a low vibration gearbox is executed with the using the Finite element shell analysis, the modal analyses and a structural optimization method in addition.

The paper gives the picture of widespread, theoretical demanding requirements and at the same time high actuality of solution of given problem. The right practical application still gained result means and in future will mean the important element of the further development of mechanical parts.

A NEW METHOD FOR IMPROVING DYNAMIC STIFFNESS OF ADVANCED CERAMIC STRUCTURES

Paul Scagnetti^{1,2}, Eric Marsh³, Alex Slocum¹, Samir Nayfeh¹

1. Precision Engineering Research Group, Massachusetts Institute of Technology

2. Precision Products Group, Wilbanks International

3. Department of Mechanical Engineering, Penn. State University

One significant disadvantage of ceramic materials when used in structural applications is the low material damping capacity. This has limited their application in precision machines where high dynamic stiffness is needed. Ceramics, such as aluminum oxide, have high static stiffness. However, the low material damping causes ceramic structures to vibrate significantly at the resonant frequencies of the structure. A new damping technology referred to as ShearDamping has increased the low dynamic stiffness of ceramics by increasing the damping. This paper will briefly review the theory of ShearDamping then focus on the advantages, difficulties, and design issues related to ShearDamping ceramic structures. ShearDamping uses a thin film of viscoelastic material backed by stiff, elastic components inside a structure. The relative motion between the embedded components and the structure shears the viscoelastic layer which dissipates vibration in the form of heat.

Structural vibration can result from energy sources such as floor noise, actuator activity, cutting forces, and movement of various components. These sources generate relatively wide band excitation that may excite one or more of the critical modes of vibration in the structure. Ceramic structures are more susceptible to vibration than cast iron or granite structures because of their relatively low material damping capacity. Although most ceramic structures have high static stiffness, they may not have high dynamic stiffness because of structural resonance. Designers concerned with vibration have often used cast iron (for heavy structures) and plastic (for light structures) because of their favorable internal damping. Hollow structures filled with other materials such as concrete may provide further vibration reduction. Constrained layer damping has been used extensively in designs which vibrate even when filled [1]. However, constrained layer techniques have limited damping capability in many applications.

ShearDamping was developed through research conducted at M.I.T. by Marsh and Slocum [2,3]. This damping technique shears a lossy material to dissipate energy, but unlike traditional constrained layer treatments, the shear medium is inside the structure. The medium is generally a viscoelastic material similar to rubber yet often only 0.1mm thick. Figure 1 shows a sketch of how the shear mechanism works. The thin viscoelastic layer placed between two smooth plates is sheared as the beams move relative to one another (because one face will be in compression and the other will be in tension). As the beams oscillate, energy is dissipated by shearing the layer.

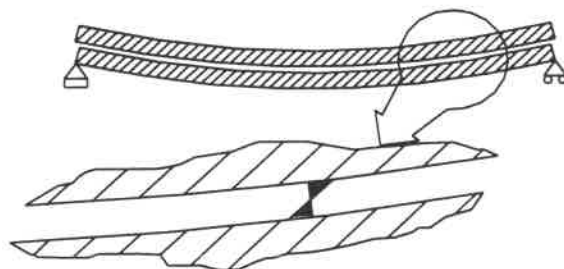


Figure 1 - Schematic of the damping layer profile in a shear damped beam.

The effectiveness of the damping in a system can be measured in terms of the vibration removed from the system per cycle. The loss factor η is the ratio of energy dissipated versus total system energy system. Thus, a system with a loss factor of 1 would not vibrate. Common steel and ceramic structures might exhibit loss factors of approximately 0.002. Filled structures might achieve a loss factor of approximately 0.01. ShearDamped steel structures with viscoelastic material have achieved a loss factor of 0.2 depending upon weight constraints and available space for shear members. The maximum loss factor for a given configuration can be estimated using the empirical expression from Marsh:

$$\eta_{theo,max} = 0.4 \left(\frac{EI_{damper}}{EI_{structure}} \right) \quad (1)$$

There are three main design issues related to ShearDamped structures. First, designing the internal damper to have maximum stiffness. As equation 1 indicates, the stiffness of the dampers is an absolute limit to the maximum loss factor. Second, the shear modulus and thickness of the damping layer must be optimized. Analytically, the loss factor is directly proportional to the shear modulus of the layer and inversely proportional to the thickness of the layer. Experiments verify this relationship except that as the layer becomes extremely thin, the shear strain goes to zero. Thus, the structure will have increased stiffness but not increased damping. Thus, the optimal layer thickness lies between the thick layer and the thin layer. Figure 2 shows the importance of choosing the correct shear layer thickness. In this plot, finite element analysis was performed on a specific configuration in order to determine which shear modulus would provide the highest loss factor [4]. For this configuration, the highest damping of the first mode occurs when the viscoelastic layer has a shear modulus of approximately 200 MPa and a thickness of 2.5mm. Third, the dampers must be designed such that the neutral axis of the dampers is not the same as the beam itself. If the neutral axes were chosen to be the same, the relative motion would be zero, the shear layer would not be displaced, and therefore no damping would occur.

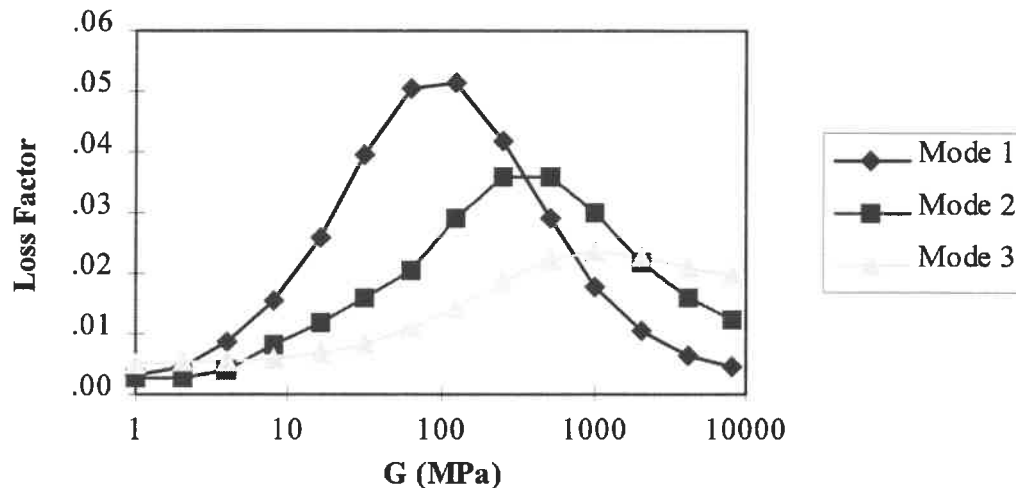


Figure 2 - Plot of loss factor versus shear modulus of the shear layer.

There are two difficult issues related to ShearDamping of ceramic structures. First, the high stiffness / weight of ceramic structures makes damping more difficult because the shear modulus of viscoelastic materials decreases at higher frequencies. In practical cases investigated, the first bending mode of a typical ceramic beam might be approximately 1500Hz. At this frequency, many viscoelastic materials have a shear modulus which is too low to be effective in damping the structure. Second, the loss factor of the ShearDamped structure depends upon the ratio of the stiffness of the internal structure to the stiffness of the main structure. With ceramic structures, it is often difficult to design internal dampers which are stiff enough to effect the structure.

FIRST PROTOTYPE: DESIGN AND EXPERIMENTAL RESULTS

This section will discuss the design of single cell ceramic beam with internal ShearDampers. The inserts were made of 96% alumina ceramic which is the same material as the beam. The components are isostatically pressed then fired, then ground to tolerance with superabrasive wheels. The inserts are assembled by wrapping the insert with damping tape, placing them inside the beam, then filling the gap between the insert and the beam with epoxy.

Figure 3 shows the prototype configuration designed for a ceramic single cell beam. To increase the damping surface while maximizing the distance between the neutral axis of the inserts and the structure, two c-shape beams are inserted in the single cell beam. Viscoelastic material is wrapped around the inserts and bonded to both along the interface at the neutral axis of the structure. The bond of both c-shapes is very important, since the potential shearing velocity in the damping layer is expected to be the highest there. The gap between the inserts and the inner wall of the single cell beam is filled with epoxy.

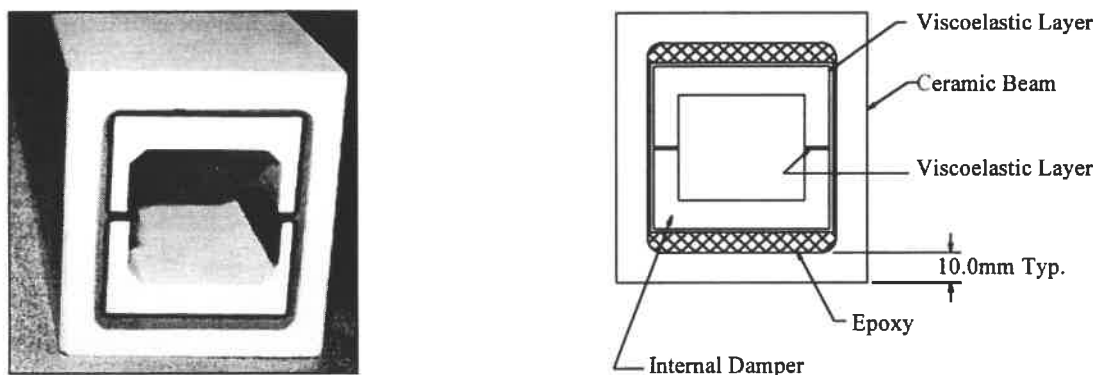


Figure 3 - ShearDamper design in single cell beam. The external dimensions of the beam are 80mm x 90mm x 712mm. The Viscoelastic layer thickness is 1mm.

The performance of the beams can be evaluated by comparing the frequency response and time response of the undamped beam to the damped beam. The time response shows the vibration of the system to an impulse force input over time. The frequency response measures the ratio of displacement to the input force as a function of frequency. The input for all results in this section was an impulse input with a hammer. The force of the hammer force was measured by a piezo-electric force transducer. The structural vibration was measured by a piezo-electric accelerometer.

Table 1 shows the time responses and frequency responses of the undamped and damped single cell beams. Both plots indicate the improvement in damping. The damping factor increased from .002 to .035. Therefore, almost all of the damping occurring in the structure is due to the energy dissipation in the damping layer. The frequency response plots also show the increase in damping and the change in natural frequency due to the inserts. The inserts add mass as well as stiffness which causes the static stiffness to weight ratio to decrease.

Table 1 - Time response and frequency response for the damped and undamped beams.

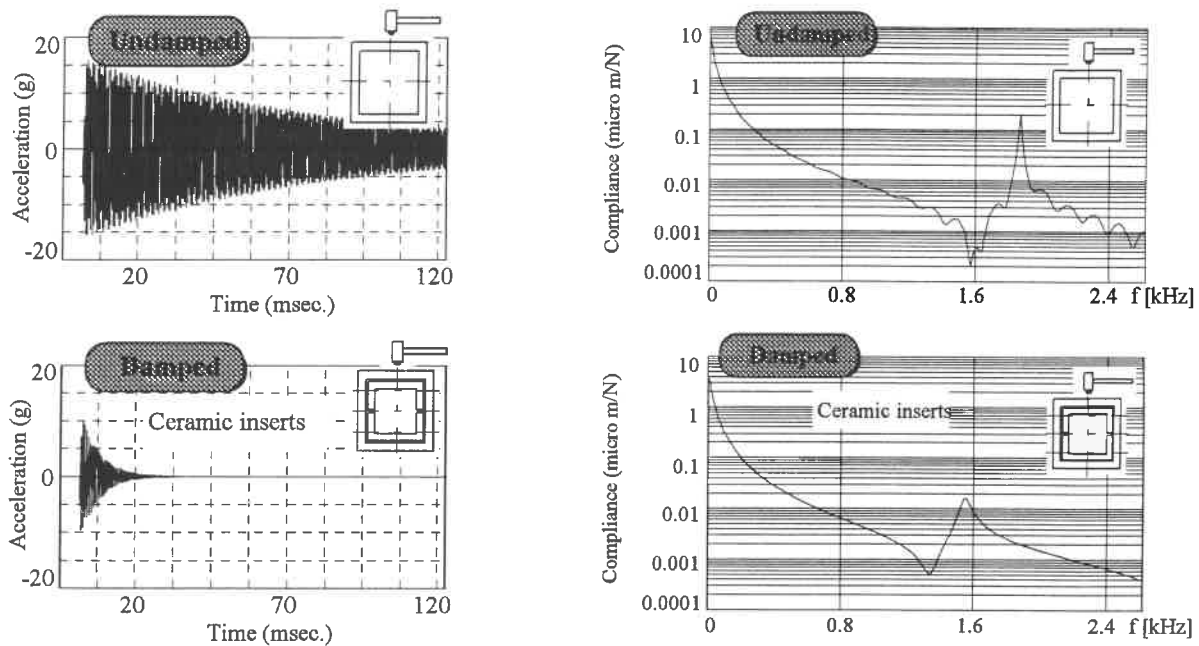


Table 2 is a summary of the main parameters for the undamped and damped beams. The difference between the theoretical loss factor $\eta_{\text{theo,max}}$ and the actual loss factor η_{exper} is due to imperfections in the manufacturing process and the modelling. The theoretical number assumes that the material is sheared everywhere which does not occur in the prototypes.

Table 2 - Characteristic parameters of the ceramic single cell beam.

Property	Symbol	Units	Undamped	Ceramic
Mass	m	kg	7.8	13.3
Elastic Modulus - Inserts	E	GPa	-	3.03
Loss Factor - Theoretical	$\eta_{\text{theo,max}}$		-	0.111
Loss Factor - Experimental	η_{exper}		0.002	0.035

SECOND PROTOTYPE: DESIGN AND EXPERIMENTAL RESULTS

Two significant changes were made to the next prototype. First, the material for the inserts was changed to a higher stiffness ceramic (Aluminum Oxide - 99.5% pure) in order to improve the stiffness ratio presented in Equation 1. The higher purity aluminum oxide has a modulus of elasticity which is 20% higher than that of the external beam but is still economical and readily available. Second, the viscoelastic layer was more precisely tuned with finite element studies in order to determine the most effective shear modulus for the shear layer. Figure 2 is an example of such optimization which can have a significant effect on the damping performance of the structure. Figure 4 is a photograph and cross-sectional drawing of the second set of damping prototype.

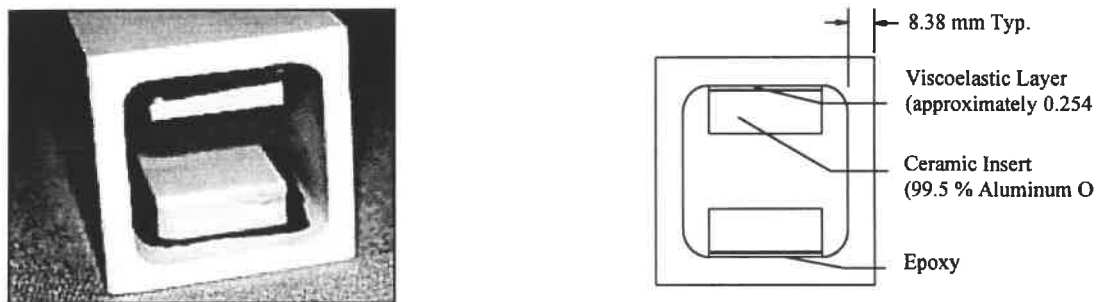


Figure 4 - Second ShearDamped ceramic beam configuration. The overall dimensions of the beam are 67mm x 67mm x 914mm. The viscoelastic layer is .25mm thick.

Table 3 shows the time response and frequency response of the undamped and damped prototype with 99.5% aluminum oxide inserts. The damping factor is increased from .002 to .02 with the ShearDampers, while the weight is increased from 6.01 to 9.07 kg. Thus, the natural frequency does not decrease as significantly as in the first prototypes.

Table 3 - Time response and frequency response for damped and undamped single cell beam.

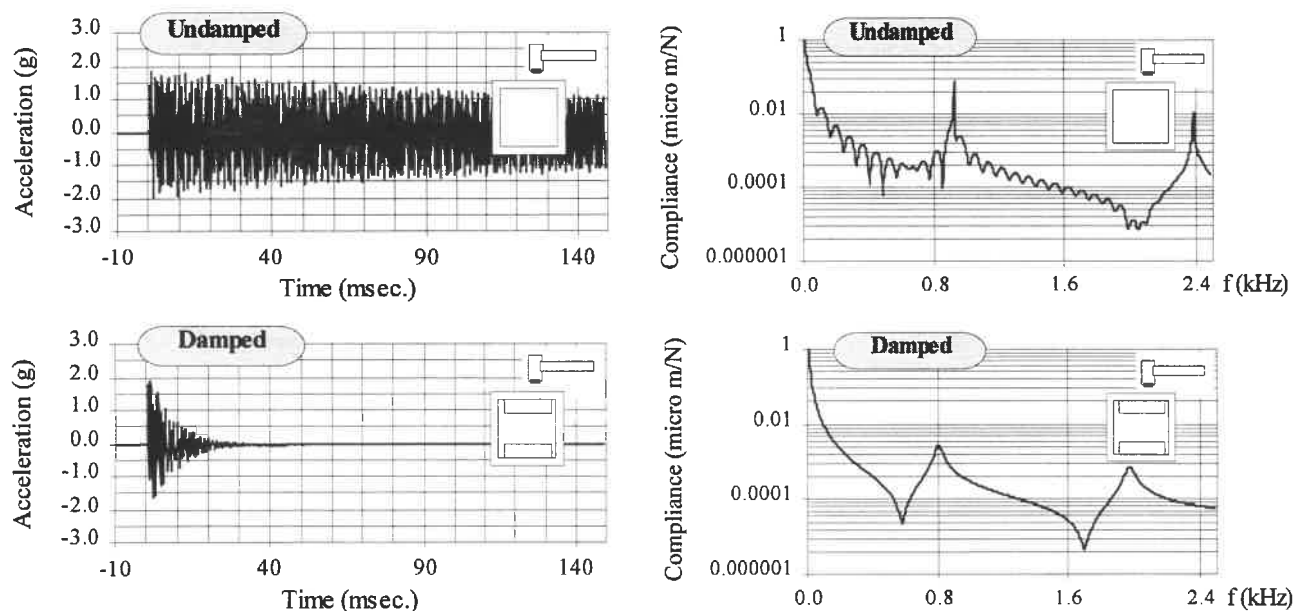


Table 4 is a summary of the main characteristics for the second prototype. The results indicate that the damping increases significantly while the additional weight of the internal dampers has been reduced. The increased stiffness of the inserts and the tuning of the shear modulus of the viscoelastic layer are the main factors in the improved performance.

Table 4 - Characteristic parameters of the ceramic single cell beam.

Property	Symbol	Units	Undamped	Ceramic
Mass	m	kg	6.01	9.07
Elastic Modulus - Inserts	E	GPa	-	372
Loss Factor - Finite Element	η_{theo}		-	0.04
Loss Factor - Experimental	η_{exper}		0.002	0.04

From these results it may be concluded that the ShearDamper technique has the potential to significantly increase the damping and dynamic stiffness of ceramic structures. Furthermore, the recent results indicate that finite element modeling of the ShearDamping technique appears to be an accurate method for predicting the performance.

The design of the damped structures should maximize the internal damper stiffness, maximize the relative motion between the structure and the dampers, and select the proper shear modulus of the damping layer.

REFERENCES

- [1] A. Nashif, D. Jones, J. Henderson, Vibration Damping, John Wiley & Sons, New York, 1985.
- [2] Marsh, E.R., "An Integrated Approach to Structural Damping", Ph.D. Thesis, Massachusetts Institute of Technology, 1994.
- [3] A.H. Slocum, E.R. Marsh, D.H. Smith, "A New Damper for Machine Tool Structures: the Replicated Internal Viscous Damper", Journal of Precision Engineering, Vol. 16, No. 3, 1994, 174-183.
- [4] C. Johnson, and D. Kienholz, "Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers", American Institute of Aeronautics and Astronautics, Vol. 20, No. 9, 1982, 1284-1290.

An Experimental Study on the High-Precision Linear Motion Table with Magnetic Bearing Suspension

Yoon Joung, Sung-Cheon Jung*, In-Bae Chang** and Dong-Chul Han***

Production Engineering Center, Samsung Electronics Co., Suwon, Korea

*Graduate Student, Dept. of Mech. Design and Production Eng., Seoul National Univ.

** Dept. of Precision Mechanical Eng., Kangwon National Univ., Chuncheon, Korea

***Dept. of Mech. Design and Production Eng., Seoul National Univ., Seoul, Korea

Abstract

The Linear motion table with magnetic bearing suspension has been developed and its performance tests were carried out. Also, the plate type capacitive transducers have been developed and adapted to the table to measure and feedback the displacement of the table. Magnetic bearings can apply electromagnetic forces to the guide rail without rigidly constraining any of 5 degrees of freedom(three rotations and two translations), thus perfectly have no friction that makes high precision possible. Furthermore, this feature allows the elimination of the complicated system compared with any others and can be easily applied in extraordinary environments like vacuum and so on. Although the table with magnetic bearing suspension is the promising technology for precision positioning field, there are many difficulties in stabilizing the 5-d.o.f table system with any control technique. The possibilities of its application in precision positioning field are presented in this paper.

1. Introduction

The essential technology in the fields of ultra precision equipments like semiconductor manufacturing equipments is the precision positioning system, which includes linear motion tables, and it mainly affects the accuracy of the whole mechanical system. According to the remarkable growth of the precision technology and growing demands on the positioning systems, many researches nowadays are under way on the development of linear motion tables.[1-4]

This paper deals with the development of magnetically levitated linear motion table system and its performance tests. The magnetic bearing system consists of electromagnetic actuator, noncontact type gap sensor and controller. The displacement measuring technology is one of the important factors for the table with magnetic suspension system and a very sensitive plate type capacitive transducer was manufactured and built in

the table. The dynamic characteristics of the levitating system including the noncontact type gap sensor are presented and the possibility of the magnetic bearing supporting linear motion table in the near future is suggested.

2. System Design

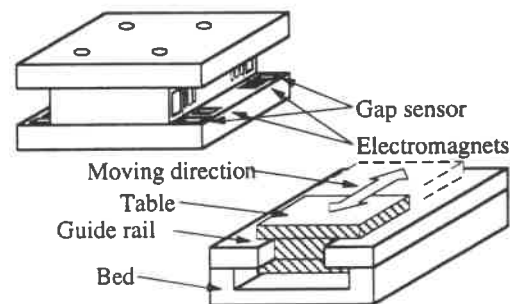


Fig. 1 Schematic diagram of the table system

Table system. The typical system of interest here consists of guide rails, a bed and a moving table with built-in gap sensors and electromagnets.

Figure 1 shows the schematics of the linear motion table system. The table, levitated by attractive electromagnetic forces, is sliding on the guide rail that is fixed to the bed. Supported by magnetic bearings, the table perfectly has no friction that needs little driving force to make sliding motion. The 5 d.o.f of table motion (heaving, serging, pitching, yawing, rolling) is controlled by 6 pairs of magnetic bearings.

Magnetic Bearing. One magnetic bearing is composed of 2 electromagnets placed opposite to each other. Figure 2 shows the electromagnet, which is to generate attractive magnetic force by the current in the coil, consists of coil and core.

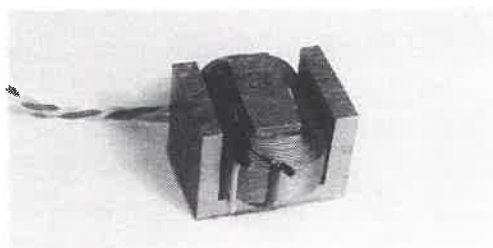


Fig.2 Coil and core of electromagnets

The specifications of the designed electromagnet are shown in table 1.

Table. 1 Specifications of the electromagnets

Nominal clearance	0.3 mm
Core	0.5mm thickness silicon steel 50 sheets
Coil	φ 0.35 enamel coated copper wire 440 turns
Area of the pole	500 mm ²

Gap sensor. The plate type capacitive transducer is the proper one that is to be used in magnetic bearing since it is not affected by magnetic field and can be placed near to the electromagnet. The capacitance between the sensor plate and guide rail decreases exponentially according to the gap, so that the signal of the sensor faced each other should be differentially amplified to improve linearization characteristics. Accordingly, the output signal voltage of the sensor and the displacement of the

table from the nominal position are related as follows;

$$V(z) = 6 \left(e^{\frac{C(g_0+z)T}{\kappa \epsilon_0 A_s \times 10^4}} - e^{\frac{C(g_0-z)T}{\kappa \epsilon_0 A_s \times 10^4}} \right) \quad (1)$$

where the nominal clearance between sensor plate and the table $g_0=0.3\text{mm}$, the sensor charging time $T=80\mu\text{sec}$, area of the sensor plate $A_s=900\text{mm}^2$, the permittivity constant $\epsilon_0=8.85 \times 10^{-12}\text{C}^2/\text{N} \cdot \text{m}^2$, permeability constant $\kappa=1$, the correction factor $C=0.03$ and z is the displacement of the table.

Sensitivity of the sensor is measured by moving object mounted on a piezo-ceramic linear table, which has $0.01\mu\text{m}$ resolution, and detecting the displacement by laser displacement sensor that has $0.05\mu\text{m}$ resolution. Measuring range was $\pm 0.05\text{mm}$, which is the linear range, using $10\mu\text{m}$ step, and the result is shown in figure 3.

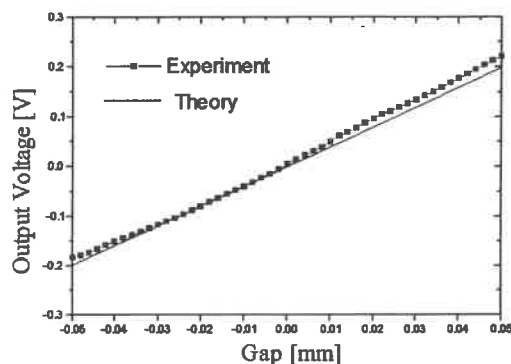


Fig.3 Displacement sensitivity of the sensor

Controller. The signals measured from the built-in sensors in the table are adjusted passing through the amplifier to have desired sensitivity and offset that all the sensors have same values. Then the signals pass analog PD compensation circuit and sum & lead network. After phase lag compensated, low pass filter suppresses the high frequency noises. Finally, the voltage signal is converted to current to actuate magnetic bearings.

3. System Modeling

Figure 4 shows the coordinate system for 5 d.o.f

Analytical Expression of Damping Ratio of Air-lubricated Slideways for Ultraprecision Machine Tools

Shigeo Kato

Nippon Institute of Technology
Miyashiro, Saitama 345, Japan

1. Introduction

Damping characteristics of air-lubricated slideways have been considered to be poor, because viscosity of the air is as small as 10^{-3} compared with the viscosity of spindle oil. Indeed, in the case of damping characteristics only by the viscosity of the air, damping ratio may be the order of 10^{-3} . Undoubtedly, this small damping ratio is inappropriate for the slideways of the hard cutting machine tools. Fortunately, it has been found that the air bearings, which are provided with restrictors, can generate a fairly large damping force which is not produced by the viscosity of the air.¹⁾ Due to author's previous researchs concerned with the frequency and the step response of an air-lubricated slideway for the ultraprecision machine tools (mass of the table: 1500 kg and static stiffness: 795 MN/m), the damping ratio equivalent to around 0.08 has been proved.^{2),3)} However, the damping ratio was only obtained by substituting numerical values for design of the air-lubricated slideways, and has not been analytically expressed.

This study analyzes the damping ratio of air-lubricated slideways. The damping characteristics of air-lubricated slideways is shown by a transfer function which has a formula of the first order in numerator and a formula of the third order of Laplace variables in the denominator. The third order denominator is analytically resolved to factors, a first order delay system with the fast decay and a second order vibration system with the

damping. The damping ratio of the second order vibration system shows the damping ratio of the slideway.

2. Equation of motion of the air-lubricated slideway

A cross-section of the air-lubricated slideway is shown in Fig. 1. A table is suspended by some sliders. Air pressure, P_s is supplied to the slider. Clearance between the slider and the guide is H . Width and length of the slider are B and L , respectively.

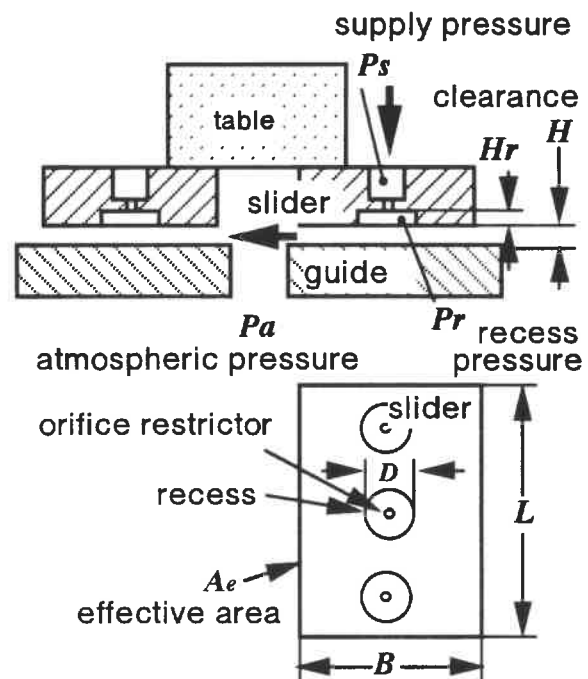


Fig. 1 Cross section of an air-lubricated slideway for ultra precision machine tools

The slider has some orifice restrictors and recesses. Diameter and depth of the recesses are D and H_r , respectively. The pressure at the recesses is P_r . An effective area of the sliders is represented by A_e .

The equation of motion of the slideway is shown as Eq. (1)².

$$m \ddot{x}(t) + K_d x(t) = f(t) \quad (1)$$

where m is the mass of the table which includes sliders of the slideway, $x(t)$ is the upward displacement of the table, and $f(t)$ is the outer force.

The dynamic stiffness K_d is represented as

$$K_d = K_s \frac{(\frac{C_2}{K_s}s + 1)}{(\frac{C_2}{K_c}s + 1)} = K_s \frac{(T_1 s + 1)}{(T_2 s + 1)} \quad (2)$$

where, s is Laplace variable, K_s is the static stiffness of the slideway and T_1 , T_2 are two time constants of the slideway. The dynamic stiffness K_d is shown in Fig. 2. In the Fig. 2, the lever ratio $l_1:l_2$ is the ratio of $K_s:K_c$.

K_s (static stiffness), K_c (compressible stiffness) and C_2 (damping coefficient) are represented as

$$K_s = A_e \left(\frac{P_s}{H} \right) \left(\frac{P_r}{P_s} \right) \frac{(b_2 - b_1)}{(a_2 - a_1)} \quad (3)$$

$$K_c = A_e \left(\frac{P_s}{H} \right) \left(\frac{P_r}{P_s} \right) \frac{\left\{ n v_c \frac{(1 + \frac{D}{B})}{2} \right\}}{\left\{ v_r + v_c \frac{(1 + \frac{D}{B})}{2} \right\}} \quad (4)$$

$$C_2 = A_e \left(\frac{P_s}{H} \right) \left(\frac{P_r}{P_s} \right) \frac{\left\{ \frac{\rho}{Q} v_c \frac{(1 + \frac{D}{B})}{2} \right\}}{(a_2 - a_1)} \quad (5)$$

Two time constants are shown as

$$T_1 = \frac{\rho}{Q} \frac{v_c}{(b_2 - b_1)} \frac{(1 + \frac{D}{B})}{2} \quad (6)$$

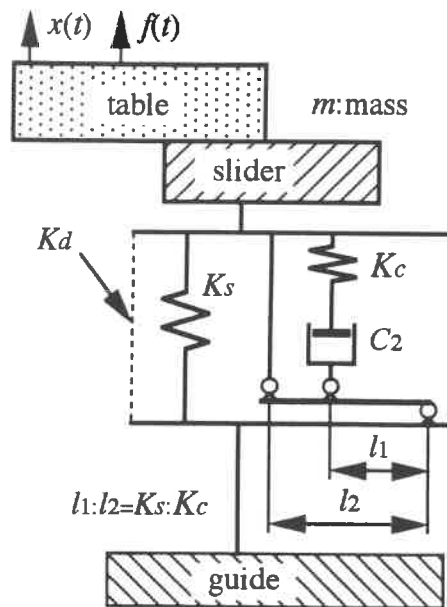


Fig. 2 The air-lubricated slideway suspended by the dynamic stiffness : K_d

$$T_2 = \frac{\rho}{Q} \frac{1}{n(a_2 - a_1)} \left\{ v_r + \frac{v_c (1 + \frac{D}{B})}{2} \right\} \quad (7)$$

where ρ is air density at the recess, Q is mass flow rate in a state of equilibrium, $v_c (=BLH)$ is the clearance volume, $v_r \{=(\pi/4)D^2H_r\}$ is the recess volume, $b_2 - b_1 = 3$, $a_2 - a_1$ is nearly equal to 3 and n is the polytropic exponent of air and is equal to 1.4.

Substituting Eqs. (2) to Eq. (1) and Laplace transforming of Eq. (1) gives

$$X(s) = \left\{ \frac{F(s)}{K_s} \right\} \frac{1}{\left(\frac{s^2}{\omega_n^2} + \frac{T_1 s + 1}{T_2 s + 1} \right)} = \left\{ \frac{F(s)}{K_s} \right\} G(s) \quad (8)$$

where $X(s)$, $F(s)$ are Laplace transformation of $x(t)$, $f(t)$, respectively, and $\omega_n^2 = K_s/m$.

$G(s)$ is a transfer function which shows the dynamic characteristics of the slideway. From Eq. (8), $G(s)$ is represented as

$$G(s) = \frac{\left(s + \frac{1}{T_2} \right) \omega_n^2}{s^3 + \frac{1}{T_2} s^2 + \frac{T_1 \omega_n^2}{T_2} s + \frac{1}{T_2} \omega_n^2} \quad (9)$$

3. Resolving to factors of the transfer function

The transfer function shown in Eq. (9) has a formula of the first order in numerator and a formula of the third order of Laplace variables in the denominator. The denominator must be solved to factors to know the physical meaning of the transfer function.

At first, the special case that is $T_1=T_2$ is considered. The transfer function is easily resolved to factors as

$$G(s) = \frac{\omega_n^2 (s + \frac{1}{T_2})}{s^3 + \frac{1}{T_2} s^2 + \omega_n^2 s + \frac{1}{T_2} \omega_n^2}$$

$$= \frac{\omega_n^2 (s + \frac{1}{T_2})}{(s + \frac{1}{T_2})(s^2 + \omega_n^2)} \quad (10)$$

In this case, the transfer function shows that it is a second order vibration system without the damping. From Eq. (10) the denominator has a factor of $\{s+(1/T_2)\}$ when T_1 is T_2 . So, when T_1 is not T_2 , the denominator is considered to have a factor of $\{s+(k/T_2)\}=\{s+(1/T_3)\}$. The new time constant T_3 shown by Eq. (11) is a time constant of the first order delay system.

$$T_3 = \frac{T_2}{k} \quad (11)$$

If $\{s+(k/T_2)\}$ is the factor, the denominator that $\{s=-(k/T_2)\}$ is substituted must be zero from the theorem of remainder. This is shown as Eq. (12).

$$(-\frac{k}{T_2})^3 + \frac{1}{T_2}(-\frac{k}{T_2})^2 + \frac{T_1}{T_2}\omega_n^2(-\frac{k}{T_2}) + \frac{1}{T_2}\omega_n^2 = 0 \quad (12)$$

Eq. (12) is rewritten as

$$\frac{T_1}{T_2}\omega_n^2 - \frac{k(1-k)}{T_2^2} - \frac{\omega_n^2}{k} = 0 \quad (13)$$

Table 1 Data for calculating the dynamic characteristics of the air-lubricated slideway

supply pressure	$P_s/P_a=6(P_s=0.6\text{MN/m}^2)$
recess pressure	$P_r/P_s = 0.75$
difference of parameters	$b_2-b_1=3, a_2-a_1=3$
density of air	$\rho=5.4225\text{ kg/m}^3$
flow rate	$Q=2.33\times 10^{-3}\text{ kg/s}$
ratio of D/B	3/100
clearance volume	$v_c=3980\times 10^{-9}\text{ m}^3$
recess volume	$v_r=94.2\times 10^{-9}\text{ m}^3$
static stiffness	$K_s=795\text{ MN/m}$
mass of table	$m=1500\text{ kg}$
effective area	$A_e=3.064\times 10^{-2}\text{ m}^2$
compressible stiffness	$K_c=1064\text{ MM/m}$
damping coefficient	$C_2=1.267\text{ MNs/m}$
time constants	$T_1=1.592\times 10^{-3}\text{ s}$ $T_2=1.189\times 10^{-3}\text{ s}$
natural frequency	$\omega_n=708\text{ rad/s}$

Dividing the denominator of the transfer function by $\{s+(k/T_2)\}$ gives

$$s^3 + \frac{1}{T_2}s^2 + \frac{T_1}{T_2}\omega_n^2 s + \frac{1}{T_2}\omega_n^2$$

$$= (s + \frac{k}{T_2})(s^2 + \frac{1-k}{T_2}s + \frac{\omega_n^2}{k})$$

$$+ [\frac{T_1}{T_2}\omega_n^2 - \frac{k(1-k)}{T_2^2} - \frac{\omega_n^2}{k}] s \quad (14)$$

In Eq. (14), the remainder is zero from Eq. (13). Thus, the denominator is resolved to factors and the transfer function is shown as

$$G(s) = \frac{\omega_n^2 (s + \frac{1}{T_2})}{(s + \frac{k}{T_2})(s^2 + \frac{1-k}{T_2}s + \frac{\omega_n^2}{k})} \quad (15)$$

where, k is a real root of

$$k^3 - k^2 - T_1 T_2 \omega_n^2 k + T_2^2 \omega_n^2 = 0 \quad (16)$$

that is a rewritten form of Eq. (13).

The part of the second order vibration system of the denominator is rewritten as

$$s^2 + \frac{1-k}{T_2}s + \frac{\omega_n^2}{k} = s^2 + 2\zeta_0\omega_0s + \omega_0^2 \quad (17)$$

From Eq. (17) the resonant frequency ω_0 and the damping ratio ζ_0 are analytically represented by k , ω_n , and T_2 as

$$\omega_0 = \left(\frac{\omega_n}{\sqrt{k}} \right) \quad (18)$$

$$\zeta_0 = \frac{\sqrt{k}(1-k)}{2T_2\omega_n} \quad (19)$$

Data of dynamic characteristics of the slideway shown in Table 1 are applied for the calculation. From values of the natural frequency ω_n and two time constants T_1 , T_2 , the value of k is obtained as

$$k = 0.853$$

Substituting the value of k into Eqs. (18) and (19), the resonant frequency and the damping ratio are obtained as

$$\omega_0 = \omega_n/\sqrt{k} = 1.08\omega_n = 788 \text{ rad/s},$$

$$\zeta_0 = 0.078.$$

The damping ratio ζ_0 is equal to the value that has been obtained by the logarithmic decrement of the step response characteristics of the slideway shown in Fig. 3.³⁾

4. Conclusions

Analytical expression of the damping ratio of air-lubricated slideways is obtained. The third order denominator of the transfer function which shows damping characteristics of air-lubricated slideways is analytically resolved to factors, a first order delay system with the fast decay (time constant is T_3) and a second order vibration system with the damping. The damping ratio of the slideway is shown by the damping ratio of the second order vibration system.

The air-lubricated slideway has two time constants T_1 and T_2 which represent the

$$\xi = \frac{1}{2\pi} \log \frac{0.710}{0.436} = 0.0776$$

$$\frac{1}{2\pi} \log \frac{0.436}{0.266} = 0.0786$$

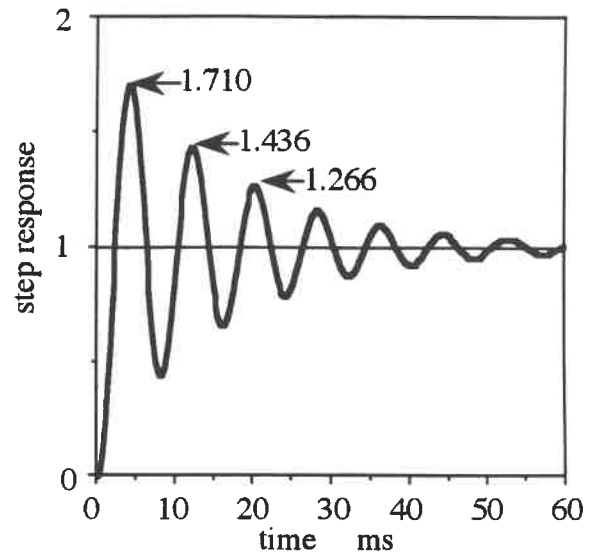


Fig. 3 Step response of the air-lubricated slideway by a unit input (calculated).

damping characteristics of the slideway. The damping ratio of the slideway ζ is expressed by the time constant T_2 , the natural frequency of the slideway ω_n , and dimensionless number k that is ratio of T_2 and T_3 as $\zeta = \sqrt{k}(1-k)/(2T_2\omega_n)$.

References

- 1) Richardson, H.H., Static and Dynamic Characteristics of Compensated gas Bearings, Trans. of ASME, Vol. 80, No. 7 (1958) 1503.
- 2) Kato, S., Damping Characteristics of Air-lubricated Slideways for Ultraprecision Machine Tools, Proc. of ASPE Spring Topical Meeting, April (1994) 63.
- 3) Kato, S., Step Characteristics of Air-lubricated Slideways for Ultraprecision Machine Tools, Proc. of ASPE Annual Meeting, (1994) 271.

MODELING OF SELF-COMPENSATED BEARINGS FOR HIGH STIFFNESS

Bi Zhang and Yunquan Sun
Department of Mechanical Engineering
Precision Manufacturing Center
University of Connecticut
Storrs, CT 06269

1. INTRODUCTION

To increase the stiffness of aerostatic bearings, both the displacement- and pressure-feedback methods are generally used. Infinite or negative stiffness can be realized under the conditions of a feedback and an energy input. In order to achieve a high or infinite stiffness, an aerostatic journal bearing was proposed by Yoshimoto that utilized a regulating ring to adjust the bearing clearances through two elastic O-rings. Loading the journal would create a pressure difference between the loading and unloading sides. The differential pressure generated a net force to balance the load applied, and to push the loaded journal back to its original position [1]. A hydrostatic thrust bearing was designed by Tully in which a laminar flow slot-restrictor was employed as the regulating mechanism to achieve a high stiffness [2]. Mizumoto used a regulating ring to achieve infinite stiffness by simply changing the operating conditions of the aerostatic thrust bearing [3]. Brzeski designed a journal bearing in which a movable bush was used for high stiffness [4]. A common feature found in the aforementioned bearings is that they use a regulating mechanism that is usually a movable or deformable element to achieve a high or infinite stiffness. The bearing clearances can be automatically regulated by this mechanism to compensate for an external load-induced displacement. Along this line, a theoretical model can be developed to direct the design of aerostatic bearings of a high or infinite stiffness.

This paper proposes a general model for high stiffness based on the principle of self-compensation. A feedback mechanism and an energy supply are the two basic requirements to realize self-compensation. Theoretical analyses are conducted on a two degree-of-freedom system with a self-compensation mechanism. This model should be useful in designing and optimizing an aerostatic or hydrostatic bearing to achieve a high or infinite stiffness.

2. PRINCIPLE OF SELF-COMPENSATION

The general model for high stiffness is established using a two degree-of-freedom linear system shown in Fig. 1. An important characteristic of the system is that no sensor or actuator is needed to realize infinite or negative stiffness. If a displacement x_1 is induced on the primary mass m_1 by an external load, it will be transmitted to the regulating mechanism m_2 through the spring k_1 . Therefore, self-sensing is realized through this transmission. The displacement of the regulating mechanism x_2 serves as a feedback to respond to the displacement x_1 , and substantially generates a reaction force proportional to the feedback to oppose the motion of the primary mass m_1 . Therefore, self-regulation is realized through the reaction force that practically is an actuating force. It should be noted that a regulating force will not be generated until the regulating mechanism has a displacement. The direction and magnitude of a regulating force are determined by the displacement of the regulating mechanism and the gain of the regulating mechanism k_1^* , respectively. The equations of motion of the system are expressed as:

$$\begin{cases} m_1 \ddot{x}_1 + c_1 \dot{x}_1 - c_1 \dot{x}_2 + k_1 x_1 + k_1^* x_2 = F(t) \\ m_2 \ddot{x}_2 + c_1 \dot{x}_2 - c_1 \dot{x}_1 + c_2 \dot{x}_2 - k_1 x_1 - k_1^* x_2 + k_2 x_2 = 0 \end{cases} \quad (1)$$

The responses of the system are obtained by taking Laplace transform of the equations:

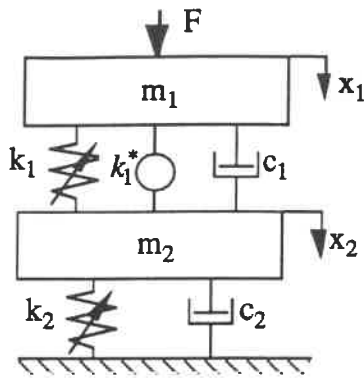


Fig. 1 Self-compensated system

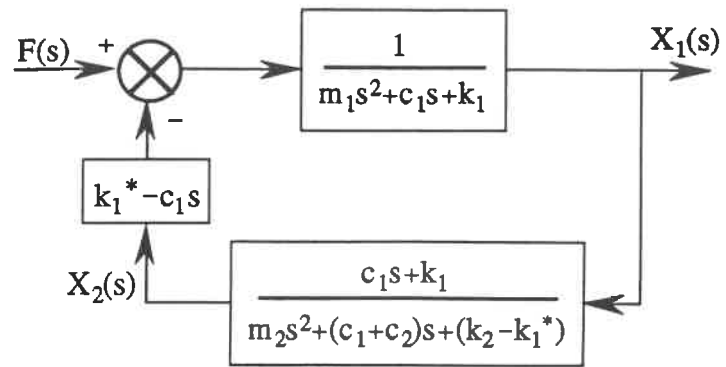


Fig. 2 Block diagram of the system

$$\begin{cases} X_1 = \frac{[m_2s^2 + (c_1 + c_2)s + (k_2 - k_1^*)]F(s)}{(m_1s^2 + c_1s + k_1)[m_2s^2 + (c_1 + c_2)s + (k_2 - k_1^*)] - (c_1s - k_1^*)(c_1s + k_1)} \\ X_2 = \frac{(c_1s + k_1)F(s)}{(m_1s^2 + c_1s + k_1)[m_2s^2 + (c_1 + c_2)s + (k_2 - k_1^*)] - (c_1s - k_1^*)(c_1s + k_1)} \end{cases} \quad (2)$$

Equation (2) shows that the steady state response of the primary mass m_1 to a step input can be positive, zero or negative, which implies that a positive, infinite or negative stiffness can be realized by an appropriate selection of the system parameters k_1 , k_2 , and k_1^* . The block diagram of the system is shown in Fig. 2. On the basis of equation (2) and the system block diagram, the following cases are presented.

$k_2 - k_1^* > 0$ for positive stiffness Under this condition, the feedback loop of the system consists of a stable second-order content. In responding to an external load, the primary mass reacts with a certain displacement in the same direction as the load applied, but never reaches zero. The regulating mechanism responds to this displacement at a relatively low level in comparing to the next cases. The system possesses a positive stiffness due to the reason that the regulating force can never be larger than the external load.

$k_2 - k_1^* = 0$ for infinite stiffness The feedback loop contains an integral content that makes the loop marginally stable. In responding to an external load, the output of the system becomes zero for the steady state response when the displacement of the regulating mechanism no longer increases but remains at a certain level due to the integral content, and the regulating mechanism responds to the zero displacement at a certain level. The force generated by the regulating mechanism is large enough to oppose any displacement of the primary mass and to balance the external load. The system stays in this equilibrium condition until the loading condition changes and infinite stiffness is realized.

$k_2 - k_1^* < 0$ for negative stiffness Under this condition, the second-order content contained in the feedback loop makes the loop unstable. In responding to an external load, the primary mass reacts with a certain displacement in the direction opposing the load applied, and the response of the regulating mechanism to this displacement is at a high level, but is bounded by the other contents of the system. The force generated by the regulating mechanism is large enough to cause the system over-responding to the external load, and pushes the primary mass back over its equilibrium position. Therefore, negative stiffness can be achieved by the unstable nature of the feedback loop. In considering the above three cases, the stability condition is found using the Routh-Hurwitz stability criterion:

$$k_1^* < \frac{(m_1 k_2 + m_2 k_1 + c_1 c_2) c_1 m_1 + (m_1 k_2 + c_1 c_2) c_2 m_1 + (m_2 k_1 + c_1 c_2) c_1 m_2}{[(c_1 + c_2) m_1 + c_1 m_2] m_1} - \frac{[(c_1 + c_2) m_1 + c_1 m_2] k_1 k_2}{(c_1 k_2 + c_2 k_1) m_1} \quad (3)$$

3. MODELING

The two degree-of-freedom system can be used to model an aerostatic bearing with a regulating ring as shown in Fig. 3(a). The pressurized air films are simulated by the springs and dampers shown in Fig. 3(b). Fig. 3(c) shows an equivalent system. The load applied to the bearing has to be limited at a level where the spring stiffnesses and damping coefficients can be treated linearly. In Fig. 3(d), the first subscript $i = 1, 2, 3$ stands for the clearances between the thrust plate and regulating ring, the regulating ring and the housing, the thrust plate and the housing, respectively. The second subscript $j = 1, 2$ represents the parameters that has a relationship with the movement of m_1 and m_2 . Pressure distribution in a bearing clearance can be determined using the Reynolds' equation:

$$\frac{1}{R} \frac{\partial}{\partial R} \left(P H^3 R \frac{\partial P}{\partial R} \right) = \sigma \frac{\partial}{\partial \tau} (P H) \quad (4)$$

$$\text{where } R = \frac{r}{r_o}; P = \frac{p}{p_a}; H = \frac{h}{h_r}; \tau = \omega t; \sigma = \frac{12\mu\omega}{p_a} \left(\frac{r_o}{h_r} \right)^2.$$

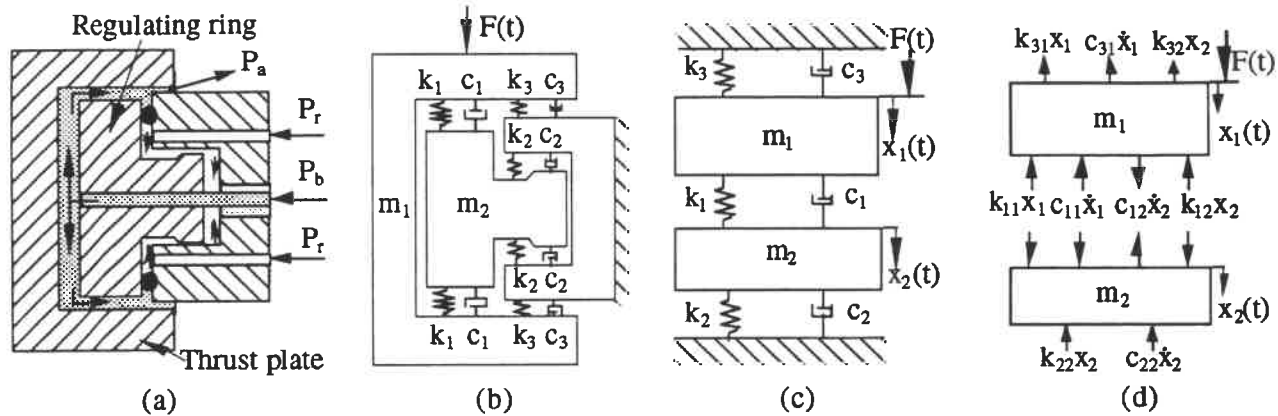


Fig. 3 Modeling a bearing with a self-compensation mechanism for infinite stiffness. (a) Aerostatic thrust bearing [3]; (b) bearing model with springs and dampers; (c) equivalent two degree-of-freedom system; (d) forces applied to the thrust plate and the regulating ring.

Perturbation technique is used to solve the Reynolds' equation [5]. By obtaining the static and dynamic pressures in the bearing clearances, the stiffness and damping coefficients are determined and the governing equation of the system can be expressed as:

$$\begin{cases} m_1 \ddot{x}_1 + c_{31} \dot{x}_1 + c_{11} \dot{x}_1 - c_{12} \dot{x}_2 + (k_{11} + k_{31}) x_1 + (k_{12} + k_{32}) x_2 = F(t) \\ m_2 \ddot{x}_2 + c_{12} \dot{x}_2 - c_{11} \dot{x}_1 + c_{21} \dot{x}_2 - k_{11} x_1 + (k_{22} - k_{12}) x_2 = 0 \end{cases} \quad (5)$$

Because the bearing shown in Fig. 3(a) requires two air supplies p_b and p_r , a different combination of these supplies results in a bearing with different characteristics. The step response of the bearing was simulated with the results shown in Fig. 4. If the condition $k_{22} - k_{12} > 0$ is satisfied, the bearing exhibits a positive stiffness. In this case, a displacement of the regulating ring would create a larger force in the regulating ring bearing than in the main bearing. The feedback loop is always stable under this condition. When under the condition of $k_{22} - k_{12} = 0$, the bearing possesses an infinite stiffness. The regulating ring oscillates in the bearing housing if the bearing is dynamically loaded. It will not stop oscillating until the steady state displacement of the thrust plate goes to zero. The

feedback loop is marginally stable, but the bearing system is stable under this condition. An integral content appears in the transfer function of the feedback loop which forms an integral feedback to eliminate the steady state error of the step response of, and thus to create an infinite stiffness for the bearing system. When the condition $k_{22} - k_{12} < 0$ is met, the bearing shows a negative stiffness. Upon a dynamic loading of the bearing, the regulating ring oscillates at a bounded magnitude in the bearing housing. The feedback loop is unstable, however, the whole bearing system is still stable under this condition.

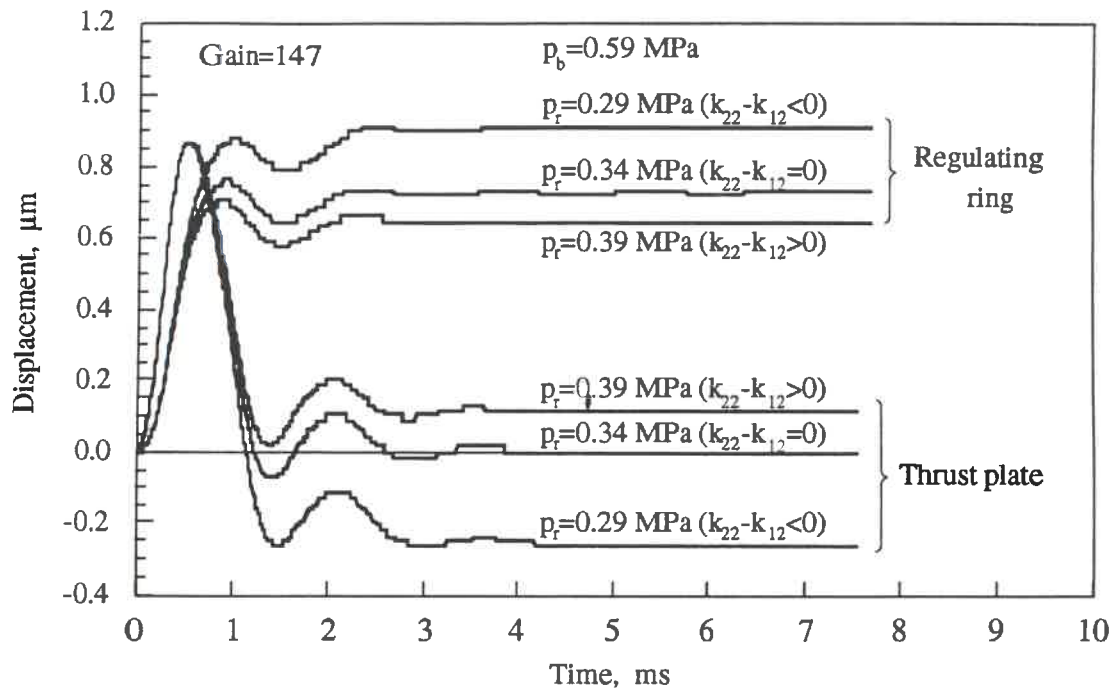


Fig. 4 Simulation results of a step response of the bearing system

4. CONCLUSIONS

A general model is proposed based on the principle of self-compensation to design aerostatic or hydrostatic bearings of a high stiffness. If the feedback loop of a bearing system is stable, the system always exhibits a positive stiffness; if marginally stable, an infinite stiffness; or if unstable, a negative stiffness. An integral content exists in the feedback loop to form an integral feedback. The integral feedback eliminates the steady state error of the system response to a step input so as to realize infinite stiffness. Application studies show that the model can be widely used in designing aerostatic or hydrostatic bearings.

REFERENCES

- 1 Yoshimoto, S. and, T. Kakubaki, The Aerostatic Spindle Using the deformable O-ring, *Journal of Japan Society Mechanical Engineers*, Vol. 52, (1986) pp. 70-78.
- 2 Tully, N., Static and Dynamic Performance of an Infinite Stiffness Hydrostatic Thrust Bearing, *ASME Journal of Lubrication Technology*, Vol. 106, (1977) pp. 106-112.
- 3 Mizumoto, H., T. Matsubara, and N. Hata, Zero-Compliance Aerostatic Bearing for An Ultra-Precision Machine, *Precision Engineering*, Vol. 12, (1990) pp. 75-80.
- 4 Brzeski, L. and Z. Kazimierski, Infinite Stiffness Gas Bearings for Precision Spindles, *Precision Engineering*, Vol. 14, (1992) pp. 105-109.
- 5 Castelli, V. and J. Pirvics, Review of Numerical Methods in Gas Bearing Film Analysis, *ASME Journal of Lubrication Technology*, (1968) pp. 777-792.

itance method is used the interpolation function for this situation (detector electrode D_2 is in its linear range of $-1 \text{ mm} \leq x_{D_2} \leq 1 \text{ mm}$) would be:

$$x_{D_2} = s \cdot (C_5 - C_2) \quad (1)$$

where s is the displacement sensitivity $\partial x / \partial C$ [m/pF]. It is easily seen that a change in s due to a distance variation can not be corrected by this algorithm without determining this distance. In principle, this is possible by using the following algorithm instead.

$$x_{D_2} = s \cdot \frac{C_5 - C_2}{C_5 + C_2} \quad (2)$$

This takes care of the problem of electrode distance dependency, but discontinuities at the switch-over points remain. This means that in this and the following algorithms the unit of s is now [m]. The following algorithm does compensate for distance variations and is continuous over the whole measurement range.

$$x_{D_2} = s \cdot \frac{C_2 - C_5}{(C_2 - C_5) - \left((C_1 - C_4) \cdot \frac{C_2 + C_5}{C_1 + C_4} \right)} \quad (3)$$

These equations are also valid for D_1 and D_3 by changing the indices appropriately. The selection of the correct capacitances to be used in the different algorithms can be implemented in a microcontroller. The influence of electrode and alignment errors on the position measurement (e.g. the angles around the X- and the Y-axis) are summarized in [4, 3].

4 Experiments and results

A laboratory test model has been designed and manufactured. In this model the electrodes are made from double-layer printed circuit board with a thickness of 2.3 mm. The double layer enables connecting the electrode elements by vias through the printed circuit board. The width of the inter-electrode gaps of the reference electrodes is 250 μm . However, the tolerances of the electrode dimensions are quite large, in the order of 50 μm . Due to the use of printed circuit board a high *absolute* accuracy and a high long term dimensional stability is not to be expected.

The effective range of this test model is 60 mm $\times$ 85 mm. The area of one electrode element is 6 $\times$ 6 mm². The distance between the sensor and matrix electrodes is 0.75 mm. This results in an essentially linear range of 2 mm for each sensor electrode. In this layout, there are 14 $\cdot$ 10 $\cdot$ 3 $\cdot$ 3 = 1260 zero points. However, for absolute measurements only (14 $\times$ 10) + (3 $\times$ 3) - 1 = 148 calibration points are needed to achieve micrometre absolute accuracy. All remaining parts of the surface of the PCB not being used as connection, matrix electrode or active sensor electrode are used as guarding. The effective capacitance to be detected is between 0 and 0.5 pF.

Measurements have been conducted using the different measurement algorithms. As it is necessary to measure the capacitance of many different electrode combinations, the measurement system based on the modified Martin-oscillator [5] was used for these measurements because it is cheap, accurate and it offers the possibility to freely select any combination of electrodes. As reference measurement system a HP 5528A laserinterferometer was used.

The resolution was 0.25 μm ; this value depends on the electronics. The long-term stability is $\pm 1 \mu\text{m}$ peak-to-peak over 60 hours. The effective measurement speed of the experimental system was about 0.5 Hz due to switching and communication time, thereby limiting practical useability. Further research is done to increase the measurement speed. Linearity is shown in table 1.

equation number	continuous	insensitive to electr. dist. var.	linearity
1	no	no	120 μm
2	no	yes	70 μm
3	yes	yes	60 μm
3 (calibrated)	yes	yes	< 5 μm

Table 1: Comparison of the different retrieving algorithms. Linearity was measured over 8 mm. Calibration of algorithm 4 was performed at 2 mm intervals.

5 Conclusions

With an improved retrieving algorithm it appears possible to significantly reduce not only the additive errors (like remaining influence of stray capacitances), as is the case with the differential capacitance method, but also multiplicative errors of which the most important one is in this case the electrode separation distance. Also the response can be made continuous at the switch-over points.

Experimental results have shown that the values for resolution (0.25 μm) and long-term stability ($\pm 1 \mu\text{m}$ peak-to-peak over 60 hours) and linearity (best value without calibration 40 μm , with calibration < 5 μm) are according to the expectations. Scale reduction of the form standard will improve the measurement resolution. It is important to note that for two coordinates only one form standard is needed.

In this experimental set-up only low-cost components are used. In future applications a relatively simple microcontroller can handle the capacitance measurement as well as the algorithm to calculate the displacements, which makes the system cheap, compact and flexible.

For absolute accuracy a full-range calibration of the sensor relative to the capacitive form standard has to be performed (max. 148 calibration points). Such a calibration can be included in production rather than in assembly because of the insensitivity to alignment errors. The flatness of the electrode plates is not critical for the same reason.

References

- [1] Botting G. *A Two-Coordinate Measuring System*. PhD thesis, University of Reading, 1972.
- [2] Bonse M H W, Zhu F, van Beek H F and Spronck J W. A new two-dimensional capacitive position sensor. *Sensors and Actuators A*, 41-42:29-32, 1994.
- [3] Jong, G W de. *Smart Capacitive Sensors: physical, geometrical and electronic aspects*. PhD thesis, Delft University of Technology, 1994.
- [4] Bonse M H W. *Capacitive position transducers; theoretical aspects and practical realization*. PhD thesis, Delft University of Technology, 1995.
- [5] Toth F N, Meijer G C M and Kerkvliet H M M. Ultra-linear, low-cost measurement system for multi-electrode pf-range capacitors. In *Proceedings of the IMTC/95*, pages 512-515, Boston, MA, April 24-26, 1995.

Extending the Accuracy and Precision of In-situ Ultrasonic Thickness Measurements

**D.E. Dunn, J.R. Cerino
Hughes Danbury Optical Systems**

During the outer surface fabrication phase of the Advanced X-Ray Astrophysical Facility, (AXAF), optics fabrication program, ultrasonic thickness gaging was used to achieve sub-micrometer measurements of material removal. AXAF will collect grazing-incidence X-rays. The eight optics are all slightly tapered Zerodur cylinders ranging in diameter from .6 to 1.2 m, with thicknesses varying from about 15 to 22.5 mm. All had an initial length of 1 meter. The inner surfaces of the mirrors are the optical surfaces of the system. Before finishing the inner surfaces, the outer surfaces were ground and polished. In addition to dimensional requirements defined by mechanical mounting considerations, the outer surface also needed to be stress-free to ensure figure stability of the important inner surface. To this end, a control-grind sequence was required. This means that each grinding step was followed by another that removed an amount of material equal to at least three times the nominal grit size used during the grinding step plus twice the accuracy of the removal measurement. This requirement held for the entire surface. The requirements of the removal measurement system were then as follows:

- 1) Cost effective to use.
- 2) Safe for the elements.
- 3) Economical to implement.
- 4) Accurate and Repeatable

The thin cylinders are large, heavy, and fragile, so handling and moving the elements had high cost both in terms of budget/schedule and risk. For this reason, in-situ measurements were attractive. However, the grinding machine environment was not an ideal one for precision measurements for several reasons: It was not isolated vibrationally or thermally, and by definition was not pristine. Reliance on any fixed reference was unattractive in that working environment.

We examined and ultimately adopted Ultrasonic Thickness Gaging (UTG) as a solution to the problem. Ultrasonic thickness gages are often used for non-destructive testing; typical precisions are thousandths of an inch or hundredths of millimeters. These instruments are time of flight measuring systems, from which thickness is inferred from an assumed acoustic velocity for the material. They use piezoelectric crystals to generate an ultrasonic pulse in the 10 - 20 MHz region, then monitor a reflected return pulse. It is well-suited to unclean and noisy environments.

The technology did have some apparent drawbacks, however:

- 1) The wavelength of 20 MHz sound in Zerodur is around 300 micrometers.
- 2) The display resolution of available units is 1 to 2 orders of magnitude too coarse.

equation number	continuous	insensitive to electr. dist. var.	linearity
1	no	no	120 μm
2	no	yes	70 μm
3	yes	yes	60 μm
3 (calibrated)	yes	yes	< 5 μm

Table 1: Comparison of the different retrieving algorithms. Linearity was measured over 8 mm. Calibration of algorithm 4 was performed at 2 mm intervals.

5 Conclusions

With an improved retrieving algorithm it appears possible to significantly reduce not only the additive errors (like remaining influence of stray capacitances), as is the case with the differential capacitance method, but also multiplicative errors of which the most important one is in this case the electrode separation distance. Also the response can be made continuous at the switch-over points.

Experimental results have shown that the values for resolution (0.25 μm) and long-term stability ($\pm 1 \mu\text{m}$ peak-to-peak over 60 hours) and linearity (best value without calibration 40 μm , with calibration < 5 μm) are according to the expectations. Scale reduction of the form standard will improve the measurement resolution. It is important to note that for two coordinates only one form standard is needed.

In this experimental set-up only low-cost components are used. In future applications a relatively simple microcontroller can handle the capacitance measurement as well as the algorithm to calculate the displacements, which makes the system cheap, compact and flexible.

For absolute accuracy a full-range calibration of the sensor relative to the capacitive form standard has to be performed (max. 148 calibration points). Such a calibration can be included in production rather than in assembly because of the insensitivity to alignment errors. The flatness of the electrode plates is not critical for the same reason.

References

- [1] Botting G. *A Two-Coordinate Measuring System*. PhD thesis, University of Reading, 1972.
- [2] Bonse M H W, Zhu F, van Beek H F and Spronck J W. A new two-dimensional capacitive position sensor. *Sensors and Actuators A*, 41-42:29-32, 1994.
- [3] Jong, G W de. *Smart Capacitive Sensors: physical, geometrical and electronic aspects*. PhD thesis, Delft University of Technology, 1994.
- [4] Bonse M H W. *Capacitive position transducers; theoretical aspects and practical realization*. PhD thesis, Delft University of Technology, 1995.
- [5] Toth F N, Meijer G C M and Kerkvliet H M M. Ultra-linear, low-cost measurement system for multi-electrode pf-range capacitors. In *Proceedings of the IMTC/95*, pages 512-515, Boston, MA, April 24-26, 1995.

Extending the Accuracy and Precision of In-situ Ultrasonic Thickness Measurements

**D.E. Dunn, J.R. Cerino
Hughes Danbury Optical Systems**

During the outer surface fabrication phase of the Advanced X-Ray Astrophysical Facility, (AXAF), optics fabrication program, ultrasonic thickness gaging was used to achieve sub-micrometer measurements of material removal. AXAF will collect grazing-incidence X-rays. The eight optics are all slightly tapered Zerodur cylinders ranging in diameter from .6 to 1.2 m, with thicknesses varying from about 15 to 22.5 mm. All had an initial length of 1 meter. The inner surfaces of the mirrors are the optical surfaces of the system. Before finishing the inner surfaces, the outer surfaces were ground and polished. In addition to dimensional requirements defined by mechanical mounting considerations, the outer surface also needed to be stress-free to ensure figure stability of the important inner surface. To this end, a control-grind sequence was required. This means that each grinding step was followed by another that removed an amount of material equal to at least three times the nominal grit size used during the grinding step plus twice the accuracy of the removal measurement. This requirement held for the entire surface. The requirements of the removal measurement system were then as follows:

- 1) Cost effective to use.
- 2) Safe for the elements.
- 3) Economical to implement.
- 4) Accurate and Repeatable

The thin cylinders are large, heavy, and fragile, so handling and moving the elements had high cost both in terms of budget/schedule and risk. For this reason, in-situ measurements were attractive. However, the grinding machine environment was not an ideal one for precision measurements for several reasons: It was not isolated vibrationally or thermally, and by definition was not pristine. Reliance on any fixed reference was unattractive in that working environment.

We examined and ultimately adopted Ultrasonic Thickness Gaging (UTG) as a solution to the problem. Ultrasonic thickness gages are often used for non-destructive testing; typical precisions are thousandths of an inch or hundredths of millimeters. These instruments are time of flight measuring systems, from which thickness is inferred from an assumed acoustic velocity for the material. They use piezoelectric crystals to generate an ultrasonic pulse in the 10 - 20 MHz region, then monitor a reflected return pulse. It is well-suited to unclean and noisy environments.

The technology did have some apparent drawbacks, however:

- 1) The wavelength of 20 MHz sound in Zerodur is around 300 micrometers.
- 2) The display resolution of available units is 1 to 2 orders of magnitude too coarse.

- 3) Since the thickness of the work piece was fixed, accuracies approaching one part in 100,000 were required.
- 4) Positioning and pointing errors could affect accuracy.
- 5) Coupling/contacting errors could affect accuracy.
- 6) The internal clock of the unit could be subject to drift at this level of use.
- 7) The internal clock of the unit we chose to work with had a resolution of 5 ns, corresponding to a thickness of about 30 micrometer.
- 8) Measuring material removed during grinding with a particular grit size necessarily means comparison of measurements made with different surface finishes, with unknown effect on acoustic reflection.
- 9) Removal data was required at a large number of sites on the optic surface.
- 10) While Zerodur has a very low CTE, sonic velocity is highly temperature dependent, to about one part in 6000 per Kelvin.
- 11) While the bulk modulus of the material was known to 1 part per 100, the acoustic properties were not well known.

The unit we chose to build around was a Krautkramer-Branson CL302. Some of the drawbacks were already addressed by that unit. The manufacturer supplied us with a listing of their code that allowed us to access internal registers to get tenth micrometer resolution. The problem of the 5 ns resolution was solved by making the circuitry that sent out the pulses completely asynchronous with the timing circuitry, so that averaging allowed sub-pulse precision without being affected by quantization errors.

Since we were interested primarily in relative accuracy between measurements of the thickness, it was only necessary to repeat the positioning, pointing, coupling and contact errors, not eliminate them. Repeatability of these errors was achieved by semi-permanently cementing eight transducers to the inside of each element and multiplexing them to the same pulser/timer unit. To a large degree, this also solved the problem of not knowing the acoustic homogeneity of the medium, as common path variations would cancel out in the computation of material removal, leaving only variability in the removed material as a source of measurement error. At the end of O.D. fabrication the probes and thermistor were removed by warming the epoxy.

The problem of the thermal sensitivity was partially solved and partially avoided. The thermal sensitivity was measured in the laboratory (figure 1), and used to correct the thickness measurements. To this end, a thermistor was colocated with each UTG transducer. The thermistors were multiplexed to a precision voltmeter. Finally, the thermal relaxation time of the medium was computed, and used to establish an athermalization time between when a fabrication process would end and the measurement cycle could commence, based on an allowable gradient between the surfaces.

A small sample of Zerodur was fabricated and instrumented with a piezoelectric probe and thermistor, just as the mirrors were. This sample was isolated and multiplexed along with the actual production pieces. In this manner, any drift in the clock would be detected, (though none was).

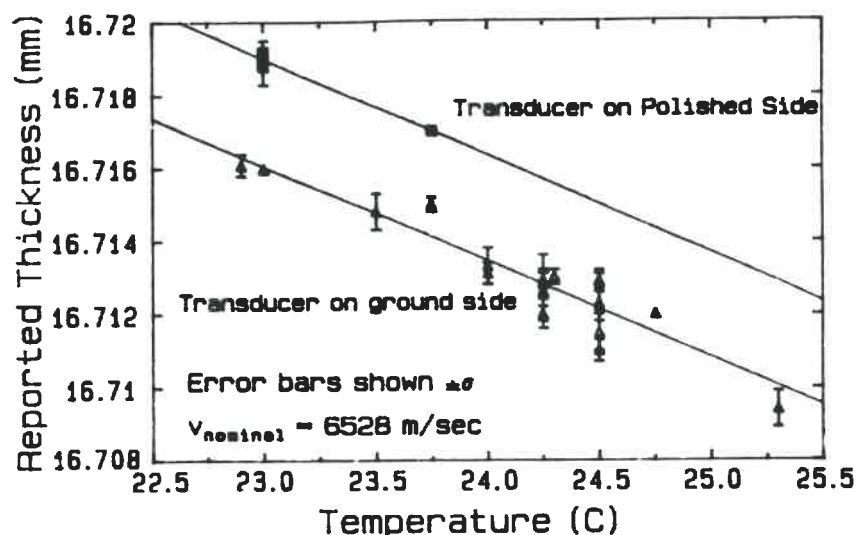


Figure 1. Speed of Ultrasonic Pulse in Zerodur varies with Temperature. $dv/dT = 0.998 \text{ m/s-K}$.

Laboratory measurements were taken to establish the effect of surface finish on ultrasonic time of flight. The goal was to establish empirically the relationship between "acoustic" thickness and "mechanical" thickness for four different surface finishes, corresponding to ground surfaces made with three different grit sizes, as well as a polished surface. The time of flight correction showed a highly linear relationship with the nominal grit size used. The magnitude of the effect was the equivalent of a few microns of material removal, as shown in figure 2.

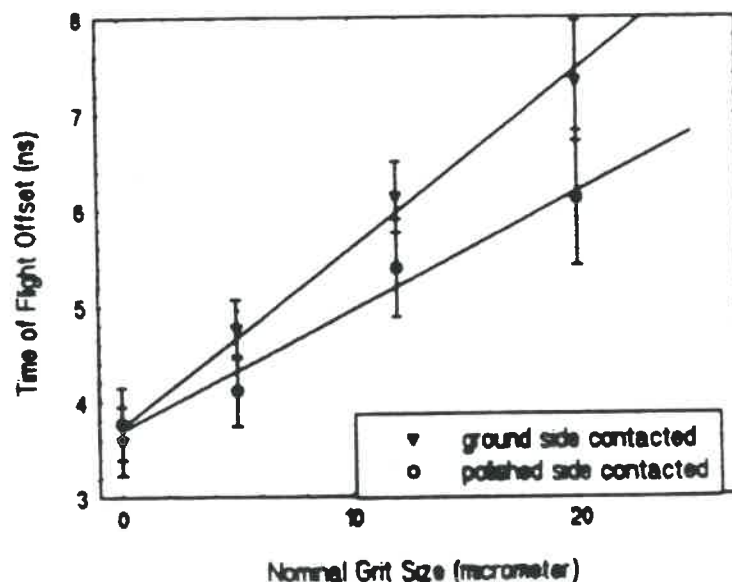


Figure 2. Dependence of Acoustic Pulse Time of Flight on Surface Finish.

The implementation directly measured removal at only eight sites. Higher data densities were available by taking axial contact profilometry scans before and after a fabrication cycle, then differencing them to establish relative removals along a line. By tying these curves at two points with absolute removal data established at the UTG sites, absolute removal data could be computed along the entire axial scan.

Finally, the expense and time of the measurement operation was reduced by automating the entire process. Instrument parameters were downloaded from a PC to the CL304 unit, the multiplexers were switched between sites by the PC, and the CL304 and voltmeter were interrogated by the PC, all automatically.

Performance

Time per eight-site measurement was approximately one hour. This consisted of 45 minutes of athermalization time, during which all set-up could be performed, and fifteen minutes to take the data.

RMS repeatability of the system was typically .1 - .2 micrometer, computed on a site by site basis. Only one optic was instrumented without material removal for a long period of time. Over seven weeks, it showed an rms repeatability of .11 micron.

As discussed above, absolute accuracy of thickness measurements were not of as much interest as absolute accuracy of the difference between measurements. Unfortunately, it is impossible to make a simple statement about the accuracy of the technique from actual in-situ use. No alternate measurements were made of removal that could rival the accuracy that can be inferred from laboratory measurements, repeatability, and an analytical consideration of the likely error sources.

The expected sources of error can be divided into three classes: non-repeatability (noise), systematic offsets, and scaling errors. No attempt was made to eliminate systematic offsets, as they are differenced out in determining removal.

Systematic scaling errors will be present in this system, but their effect on removal data can be limited by verifying the actual thickness of the optics measured. For example, a contribution of no more than 1% error on the amount of material removed (15 - 20 micron in polish) could be ensured by verifying that the actual thickness based on the time of flight measurement agreed with the mechanically measured thickness to within 1%, an easily verified .15 - .20 mm. With the exception of potential local variations in material properties, all other errors that affect acoustic velocity should show up as noise.

Non-repeatability errors need to take into account not only instrument errors, but variability induced by the fabrication processes. A worst case level of performance can be established by looking at the long-term thickness evolution. Figures 3a and 3b show the evolution at eight sites during polish-out for the P6 optic. The extent to which these lines increase predictably and smoothly is taken to be an indication of the repeatability of the

process. The rms deviation from a straight line fit to these removals is .51 micrometer for the case of optic P6, and .85 micrometer for P3. These fluctuations are a worst case measurement of the system noise, as some is due to accurately measured variability in the polishing rate of the optics under intentionally and unintentionally changing conditions. Since systematic effects can be argued to contribute negligibly to the accuracy of the measurement, we believe that the primary source of removal measurement error is the repeatability of the system. As a lower bound, the repeatability of the system on optics that aren't being polished (.1 - .2 micrometer, 1-sigma) and an upper bound established by the fluctuation about the linear removal rate (.5 - .8 micrometer, 1-sigma).

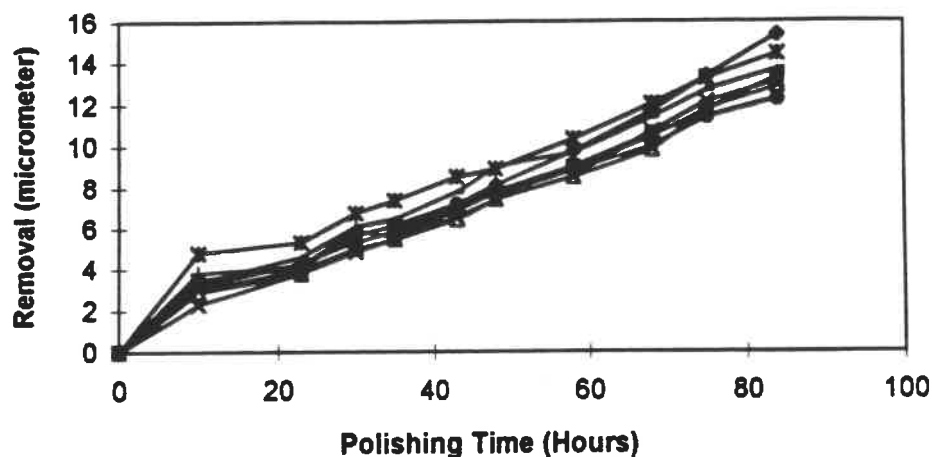


Figure 3a. Material Removal During Polish, measured at 8 sites for the P6 Optic.

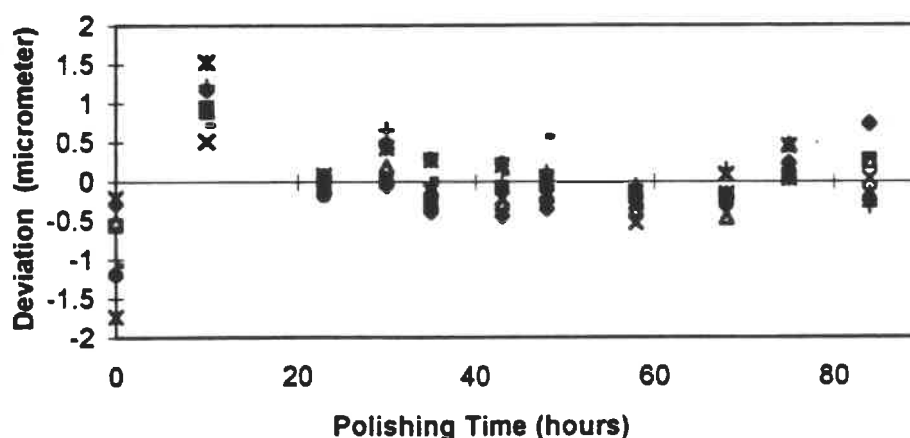


Figure 3b. Material removal during polish, measured at 8 sites for the P6 Optic. Deviation from Best Fit Line

Conclusions

Most of the problems of monitoring removal ultrasonically can be mitigated.

Ultrasonic thickness gaging is most appropriate when used to study relative thickness (removal).

The technology is well-suited for in-situ application.

It is our belief that sub-micrometer (1-sigma) accuracies can be realized over thicknesses of a few centimeters in a shop environment.

Notes and Acknowledgments

The authors wish to acknowledge and thank Beverly J. Fergason, who performed the long term repeatability analysis.

Inclusion of the manufacturer and model of the base UTG unit is for completeness; and does not represent HDOS endorsement.

Development of the UTG system was supported by TRW subcontract DS0508NL9S for NASA Marshall Space Flight Center.

Evaluation of High Precision Triangulation Sensors for Coordinate Measurement

Mysore Ananda, Jeffrey Jalkio, and Larry Konicek
CyberOptics Corporation, 2505 Kennedy St. NE, Minneapolis, MN 55413

Introduction

The sensors used in this research are based on the well known principle of laser triangulation. In laser triangulation (Figure A), a spot of light is projected from the sensor to the surface of the object to be measured. Light reflected from the surface is imaged along a different optical axis onto an optical detector. The position of the spot's image on the detector varies with the object distance and the object distance can be calculated from the position of the spot's image. Laser triangulation sensors are used for a variety of gaging applications in many industrial settings. They are increasingly being evaluated for use on general purpose coordinate measuring machines. However, these sensors are subject to radically different sources of error than are contact probes. Previous research[2] has evaluated the fundamental accuracy limits of such probes and these results indicate that they are theoretically capable of precise measurement. A. Garces et al.[1], have presented a methodology for evaluating optical coordinate measuring probes. In their report they have described various error sources contributing to the performance of optical coordinate measuring sensors. They have also described the effects of these error sources on the sensor measurement results and test procedures for performance evaluation. In this paper we will examine the causes of measurement error in several types of laser triangulation sensors and how these error sources affect sensor performance. We use the static sphere test described in [1] to compare triangulation sensors using area arrays, linear arrays, and lateral effect photodiodes (analog) as detectors. We also evaluate the theoretical explanations of these effects presented by A.

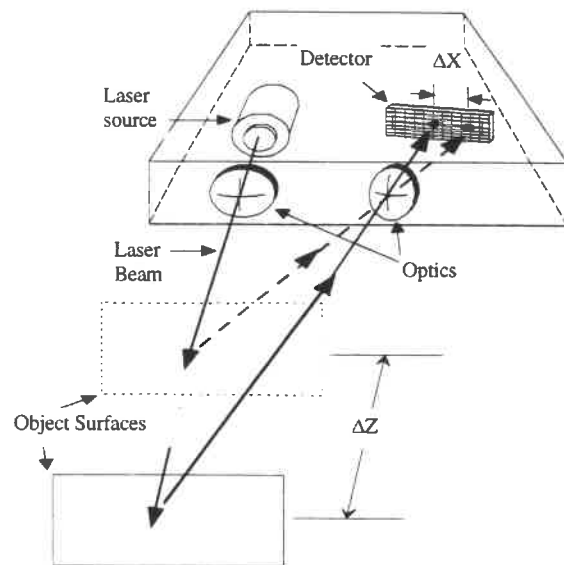


Figure A.

Garces et al., and offer some alternative explanations that account for both the their findings and our own.

Sphere Tests

The Static Sphere Test examines the effects of varying slope, curvature, surface reflectivities, volume scattering and specular reflection on an optical measuring probe. In our experiment (Figure B) triangulation sensor was made to follow the trajectory of the sphere by keeping the object-sensor distance a constant. The points on the sphere were measured at 1° angular increments starting at the pole. The tests were carried out in both the triangulation plane (X-Z) and the plane perpendicular to the triangulation plane (Y-Z). At each position on the sphere the XYZ coordinates of the sensor were recorded. The test was conducted on all three sensors (Table 1) and

with spheres of different materials. A single least square fit is made to the data and the actual radius of the sphere was computed. Graph of the deviation of the measured radius of the sphere from the actual radius at each position on the sphere indicates effects of curvature, reflectivities and other error sources on measurements using a triangulation sensor.

Sensor Detector Technology	Stated Resolution in μm	Triangulation Angle	Stated Range in mm
Linear Array Detector	0.75	35°	0.75
Area Array Detector	2	30°	0.81
Analog Detector	0.2	35°	± 3.0

Table 1.

Analysis of experimental data

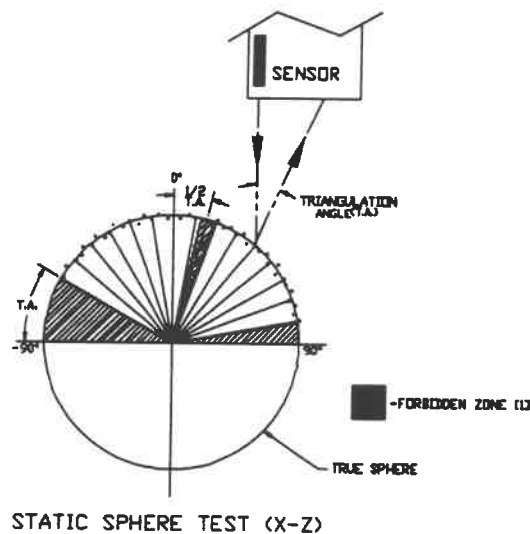


Figure B.

We observe that the X-Z plane data (Figure C) falls into three regimes. Errors in each regime are dominated by different effects. The magnitude of these errors as well as the size and location of the regions provide complementary information about the range sensor. First, where the sphere occludes receive aperture or at the sphere's equator, the laser spot cannot be observed, and no valid height readings can be obtained. Near these occlusion regions, the laser spot is partially occluded, resulting in lateral shifts in apparent spot position, hence errors in the measured height. Second, near the angle of specular reflection, height readings vary widely as the specular glint traverses the receive aperture. Finally, over the remainder of the sphere, diffuse scattering from the surface is the main source of light.

Diffuse regions

We find that outside the "forbidden regions" due to occlusion and specular reflection, the root mean square errors of these sensors varies little with angle. In these regions, performance is determined by surface microstructure and volumetric scattering (translucence).

Occlusion regions

Clearly, no height values should be reported when the laser spot is occluded by the sphere. Any signal from the receiver under these conditions is due to one of four effects. The sensor may use memory to avoid data drop out during scanning. If possible this feature should be disabled for testing. The sensor could be sensitive to ambient lighting. The detector electronics may introduce electrical noise that produces random fluctuations in the output signal. Finally, light from the

sensor may reach the receiver through circuitous paths. For example, light scattered from the source optics might be reflected from a specular sphere, or light penetrating a translucent sphere might scatter through the interior of the sphere to points on the surface visible to the receive optics. In any case, seemingly "valid" height measurements from occluded regions indicate excessive sensitivity to "noise", although the source of the noise may differ from case to case.

Specular regions

It has been suggested that specular glints cause range errors only for aberrated optical systems. Although lens aberration can result in the behavior observed, our theoretical analysis shows that these phenomena occur even in idealized systems with finite spot size. The reflected intensity is the product of the incident intensity and the local reflectivity. As shown in figure E, when the incident spot center is near the point where the specular reflection passes the receive aperture, the reflected spot profile is highly asymmetric due to the asymmetry of the reflectivity. For an ideal sensor this results in the error plot shown in figure D. The magnitude of this effect is determined by the diameter of the sphere, the size of the laser spot, and the relative intensity of the specular and diffuse spots. The range of angles over which the specular glint affects height readings is dependent on the spot size and the numerical aperture of the receive optics.

Other effects

The analog sensor tested produced some height measurements that did not correlate well with the true profile of the sphere. This particular sensor was designed for factory use and had a transparent plastic cover over the source optics to seal the sensor and prevent damage to the laser focusing lens. Unfortunately, the plastic cover scattered light from its surfaces. This scattered light was reflected by the metal spheres into the receive aperture. When this spurious light was blocked, the sensor readings changed by as much as 300 microns. We have not observed this effect in sensors designed with anti-reflective coatings on the source optics.

Conclusion

The results from the static sphere tests indicated that the performance of linear and area array triangulation sensors were much better than that of analog triangulation sensor tested. This could be due to the detector technologies, processing algorithms and the optical design. Further tests are required to determine the sources of these differences. The simulations and the experimental data from the static sphere tests prove that the specular reflection is observed even in well corrected (small lens aberration) systems and hence is inherent of any triangulation sensor.

References

1. A. Garces et al., "Performance Test Procedures for Optical Coordinate Measuring Probes", Final Project Report Part 1, European Communities Brussels, Contract No. 3319/1/0/159/89/8-BCR-D (30), July 1993.
2. Rainer G. Dorsch et al., "Laser triangulation: fundamental uncertainty in distance measurement", Applied Optics, Vol.33 No. 7, P1306-1314, March 1994.

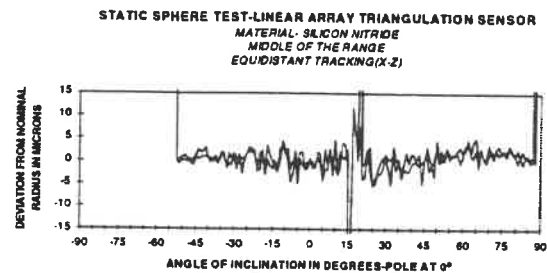
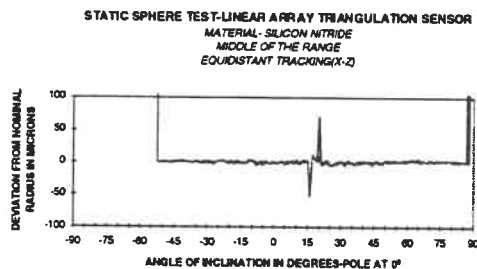
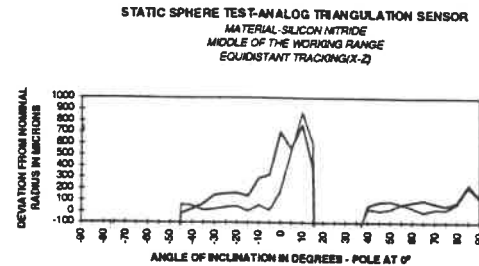
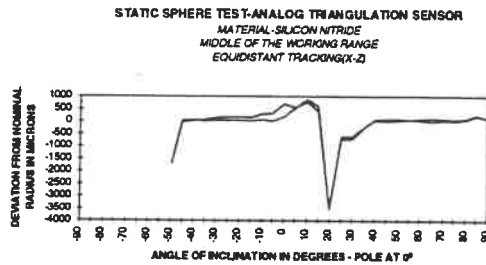
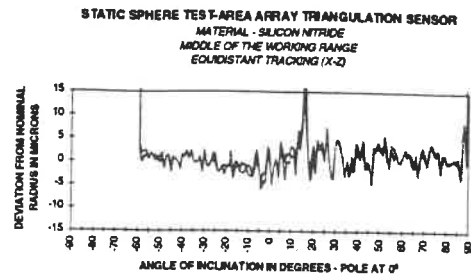
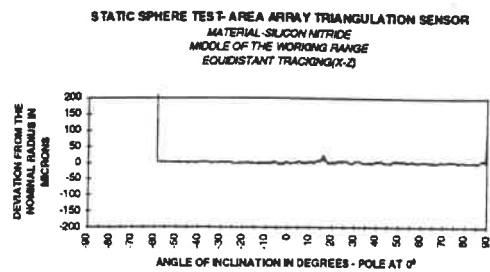


Figure C.

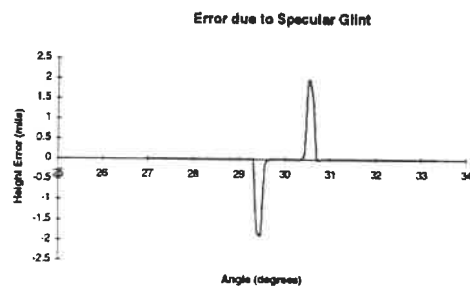


Figure D.

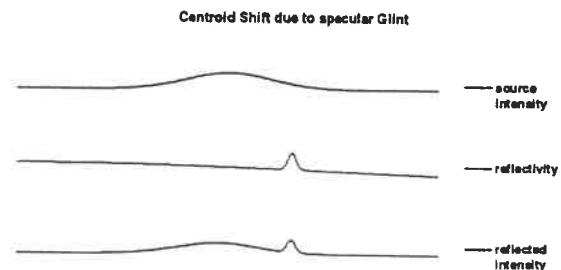


Figure E.

MEASUREMENT OF FRICTION AT LIGHT LOADS IN POLYPYRROLE THIN FILM BEARINGS

X. Liu, D. G. Chetwynd, J. W. Gardner and S. T. Smith¹

Centre for Nanotechnology and Microengineering, University of Warwick,
Coventry CV4 7AL, UK.

C. Beriet and P. N. Bartlett

Department of Chemistry, University of Southampton, Southampton SO17 1BJ, UK.

INTRODUCTION. Nanotechnology and metrology are creating ever increasing demand for super-precision translations which require sliding between their component parts. Thin film polymeric bearings sliding dry against optically polished glass datums have been used in high precision linear and angular movements over a large range, giving low and smooth friction [1,2]. Conventionally, a PTFE-lead composite contained within a sintered bronze matrix is used for the bearing as its thermal conductivity and stiffness are almost unaffected by the PTFE. Recent advances in molecular engineering have lead to organic semiconducting materials, including electroactive conducting polymers, produced by the electrochemical polymerisation of suitable monomers in either aqueous or non-aqueous solutions. Such a deposition process is well-controlled and works on irregular surfaces, suggesting their application to a wide range of situations where PTFE is difficult to use. The choice of monomer, counter-ion, solvent, and growth potential determines the film morphology and can lead to smooth, fibrillar or spherical microstructures. This controlled diversity should enable polymer coatings to be optimized to a particular tribological requirement. A feasibility study has shown that with suitable designs a pyrrole-based film could have a friction coefficient comparable or even less than that of PTFE [3,4]. A systematic study is now being carried out with improved instrumentation and using a range of surface analysis techniques. Poly(pyrrole) has been chosen because it gives a wide range of friction coefficients under different deposition conditions. Attempts have been made to describe and understand the statistical relationships between the deposition process, the topographical structure and the tribological performance of poly(pyrrole) film.

TEST APPARATUS AND INSTRUMENTATION. A special test-rig has been designed, Figure 1, consisting of a rotating optically-flat glass disk and a sample assembly to hold the polymer specimen and to carry a height probe and a parallel flexure monitored by high precision capacitive micrometry for friction measurement. The disk is driven from above, with the specimen pressed against its underside by means of a ball-bearing supported lever and dead weight loading. The friction test-rig can be operated at a speed from 0.1 to 120 mm s⁻¹ with normal loading from 0.1 to 5 N. The vertical capacitive probe is built from a Zerodur base with an etched step, around 20 µm deep, on which an aluminium electrode is deposited. A glass cantilever beam of dimensions 8.5 by 1.5 mm by 100 µm thick, aluminium coated on one side to form the other electrode, is mounted over the step. A 1 mm sapphire ball is glued onto the free end of the cantilever beam to

¹ Now at Precision Engineering Laboratory, University of North Carolina-Charlotte, Charlotte NC 28223, USA.

form a contact tip. The probe height on the stage can be adjusted by a screw-driven wedge to contact the disk with a force in the micronewton range. Once in position the probe can be decoupled from the screw to improve thermal stability. The vertical height variation, and thus the wear-rate, can be monitored continually by the probe which has nanometre resolution. Run-out in the disk bearings causes a small vertical oscillation of the whole stage and a central line is extracted to be used as height variation. Virtually the whole specimen stage is an aluminium notch-hinge simple flexure mechanism which is deflected slightly by the drag force of the disk on the specimen. Friction is measured by another capacitive gauge. A flat alumina tile with a deposited gold-film electrode is glued onto the side wall of the moving platform of the flexure, while the other electrode is a gold coated glass lens glued onto an aluminium rod that is in turn clamped to the base of the flexure. Using a spherical shape reduces tilt sensitivity and allows the electrode to be repositioned without the need for recalibration. Both probes are measured by precision capacitive bridges (Queensgate Instruments Ltd. UK). The sensitivity can be adjusted by varying the nominal gap distance of the capacitive probe to allow a reasonable linearity and resolution to 10 μN is routine. The non-linearity is kept below 1% of the range used for the friction probe and 0.1% for the height probe.

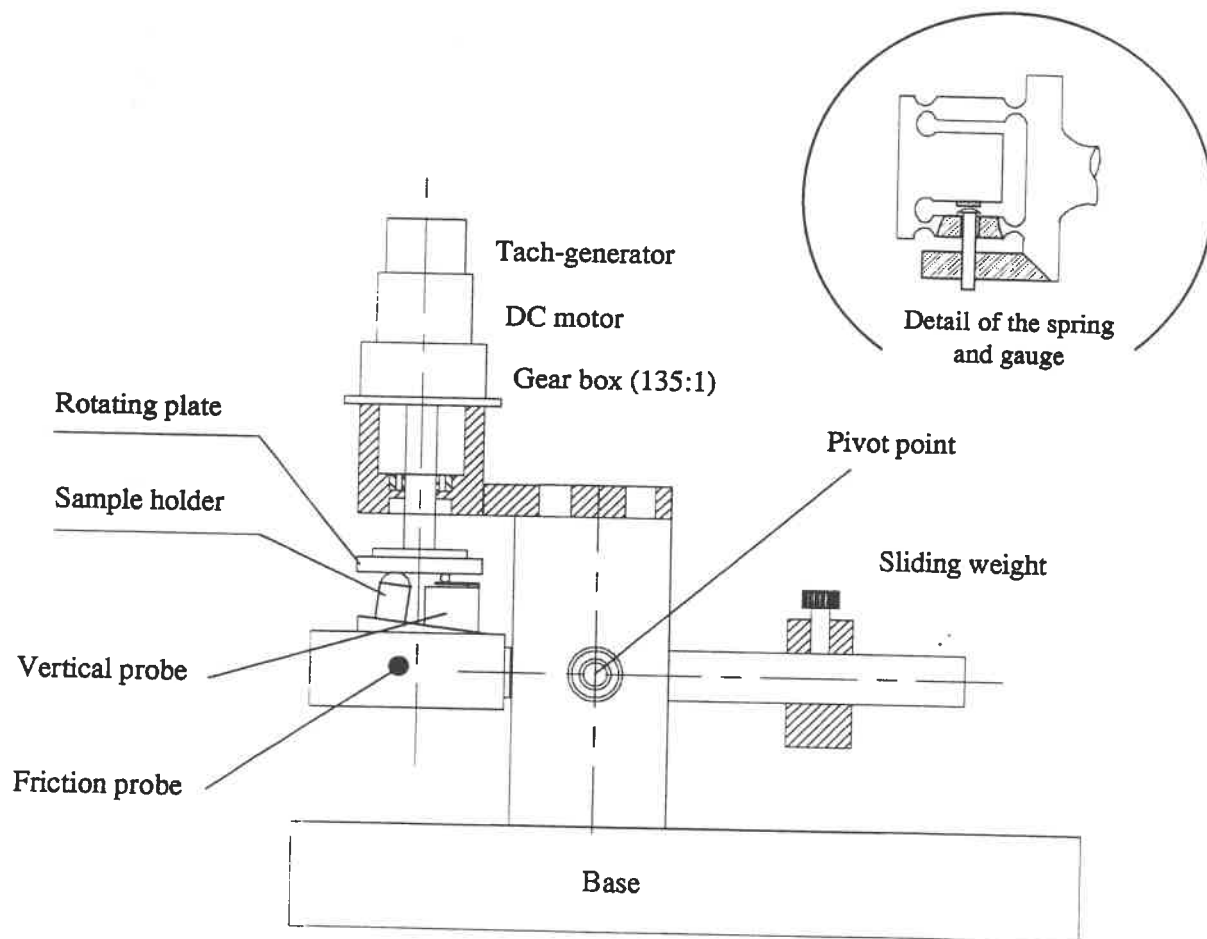


Figure 1 General arrangement of the friction and wear test-rig.

SPECIMEN SURFACE ANALYSIS. The bearing specimen consists of a plano-convex glass lens with a radius 21 mm and a thickness of 3 mm. Four 100 nm thick gold electrodes are vacuum coated onto its spherical surface and the polymer is electrodeposited onto these. Producing a set of four identical films on a single lens reduces the batch process variability and improves the study of post-deposition conditioning such as a low temperature bake. Over 70 specimens were produced from polypyrrole using different counter-ions, deposition potential, and film thickness. Specimens have been characterised by both AFM and SEM before and after friction testing in order to relate the surface morphology to tribological properties. The AFM was a special, large system with good metrology designed for convenience in engineering applications [5]. The measurements reveal that poly(pyrrole) films with different counter ions have surfaces of typical spheritic or granular patterns, with the size of spheres ranging from 0.2 to 5 μm . The thicker the film the larger size of the polymer agglomerations. They can be sparsely scattered across the specimen surface, relatively evenly packed, or clumped together to form even bigger lumpy spots. The structure seems mainly related to film thickness (below 1 μm) and the relative hydrophobicity of the polymer and substrate.

TRIBOLOGICAL RESULTS AND DISCUSSION. The main sequence of tests has confirmed and added confidence to the results of the pilot study. Tribological performance of poly(pyrrole) films varies with film thickness and microstructure and is determined in part by the choice of counter-ion. Results are reproducible which indicates good experimental control. Figure 2 shows the range of friction coefficient values observed from poly(pyrrole) (PP) with five counter ions: methane phosphonic acid (MPA); butane phosphonic acid (BPA); decane phosphonic acid (DPA); toluene sulphonic acid (TSA); and dodecyl benzene sulphonic acid (DBSA). Friction coefficients in the range 0.05 to 0.1 (similar to PTFE) are observed with some pyrrole-based films. In general, friction coefficients vary widely with PP/DBSA, PP/TSA and PP/BPA groups but only modestly with PP/DPA and PP/MPA, so that property of the latter films can be more easily controlled. The wear rates of the specimens are generally a few nanometres per millimetre sliding, some, particularly of PP/TSA and PP/DPA, offer a rate close to or slightly better than that of PTFE. With sub-micrometre films on smooth substrates even these low rates mean that bearings wear out over quite small total distances, so they are perhaps more applicable to micromechanical systems than to conventional slideways (where PTFE or lubricated acetal can normally be applied and so remain the obvious choice). An ideal bearing material should have consistent behaviour under changes of load and speed. The friction coefficients for a typical PP/MPA specimen under variations of load from 0.1 to 5 N and speed from 0 to 28 mm s^{-1} are plotted in Figures 3 and 4. In Figure 3 'theoretical' indicates a regression of the data to $L^{-1/3}$, as Hertzian theory would predict, while the 'best fit' is to the general least squares form L^b ($b=-0.93$ for PP/MPA and -1.025 for PP/TSA). The reason for this nearly inverse proportionality is not clear and is being further investigated. The friction coefficient tends to change with speed, significantly so at lower speed, but remained usefully small throughout the test range.

ACKNOWLEDGEMENT. This work has been generously supported by the UK Engineering and Physical Science Research Council under grant GR/H36382.

REFERENCES

1. Lindsey K, Smith ST and Robbie, CJ (1988), *Ann. CIRP*, **37**, 519-522.
2. Smith ST, Harb S and Chetwynd DG (1992), *J. Phys. D.*, **25** (1A), 240-248.
3. Smith ST, Harb S, Eastwick-Field V, Yao ZQ, Bartlett PN, Chetwynd DG and Gardner JW (1993), *Wear*, **169**, 43-57.
4. Gardner JW, Chetwynd DG, Smith ST, Harb S, Yao ZQ, Bartlett PN and Eastwick-Field V (1994), *Sensors and Actuators A*, **41-42**, 300-303.
5. Xu Y, Smith ST, Atherton PD, Judge T and Jones R (1994), *Proceedings of ASPE 9th annual meeting*, Cincinnati, USA, 23-28.

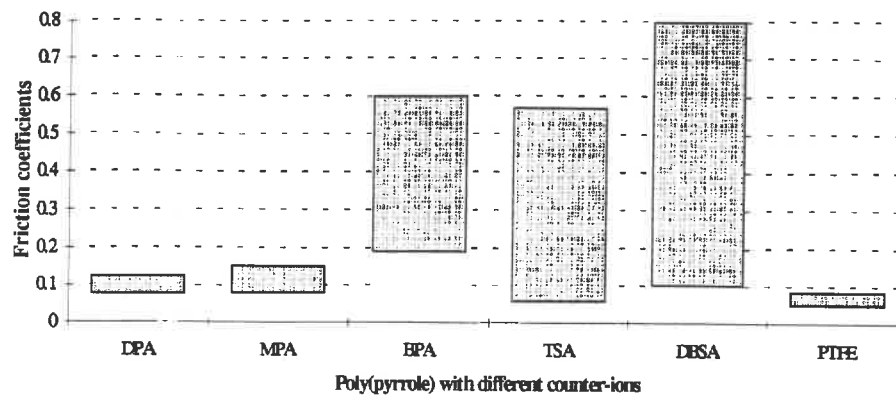


Figure 2 Range of achievable friction coefficient for the poly(pyrrole) specimens.

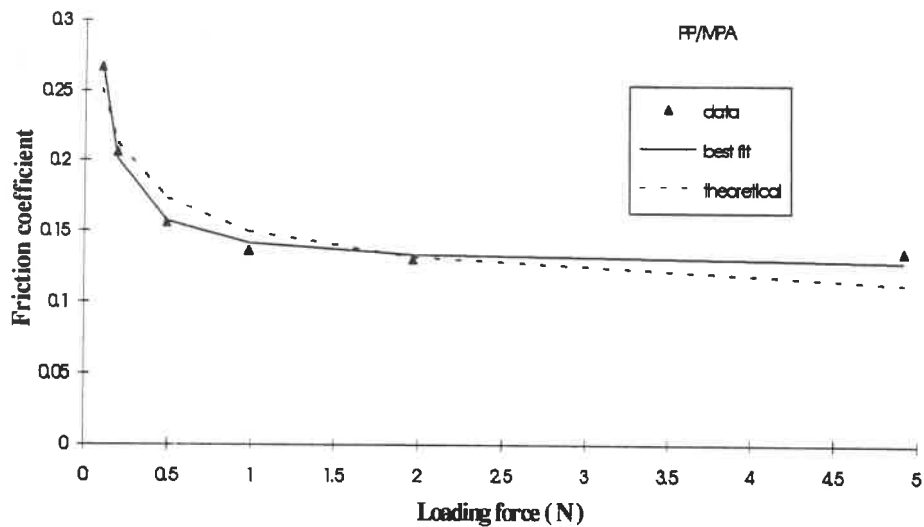


Figure 3 Effect of load on friction coefficient for PP/MPA bearing.

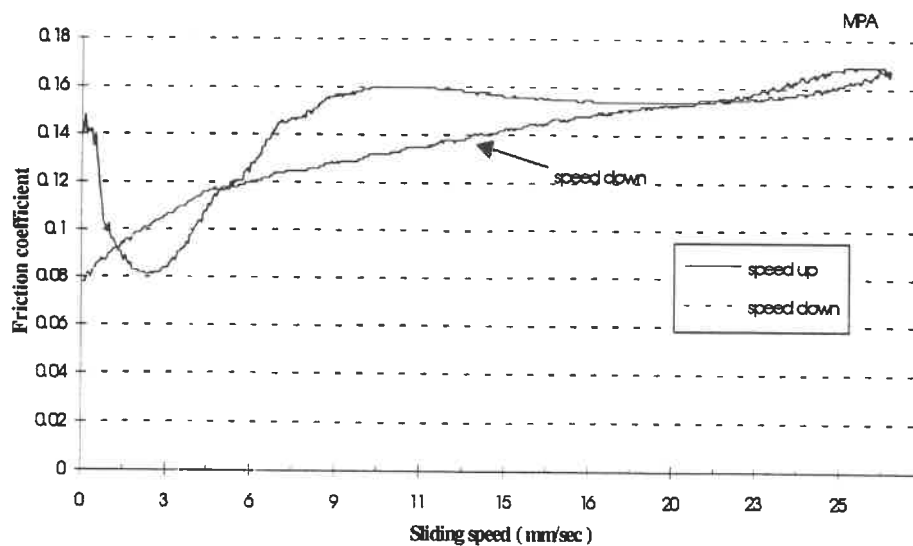


Figure 4 Effect of sliding speed on friction coefficient for PP/MPA.

An Architecture for Agile Assembly

Ralph L. Hollis and Arthur Quaid

The Robotics Institute
Carnegie Mellon University
Pittsburgh, Pennsylvania, USA

Abstract

This short paper outlines new hardware and software technologies and methods being developed for automated assembly of precision high-value products such as magnetic storage devices, palmtop and wearable computers and other high-density equipment. Our Agile Assembly Architecture (AAA) supports the creation of miniature assembly factories (minifactories) built from small modular robotic components. The goals are to substantially reduce design and deployment times and product changeover times, greatly improve quality levels by using sensor-moderated precision motion, and to achieve new levels of manufacturing system portability by shrinking the sizes of typical assembly systems from large room size to tabletop size.

We are developing the key electromechanical elements including novel subproduct couriers, manipulator/feeders, and other modular components. We are implementing a distributed realtime computer architecture, modeling and simulation software, high-level network communication protocol, and graphical programming tools to support this architecture. "Real" minifactories will be programmed by directly interfacing with automatically synchronized "virtual" minifactories.

1 Introduction

Automated assembly is a key part of our manufacturing capability. Current assembly systems have a number of shortcomings such as costly and time-consuming changeover, parts feeding problems, barely adequate inherent precision, lack of programmability and flexibility in horizontal conveyances, limited robot workspaces and large dedicated cleanroom floor space.

In this paper, we outline a new project to develop highly agile automated assembly systems which support geographically distributed design and deployment, and which can be rapidly reconfigured to meet changing market opportunities.

The SCARA assembly robot

A large percentage of today's automated assembly systems are pipelined configurations of SCARA (Selective Compliance Assembly Robot Arm) manipulators and product conveyor belts. The SCARA was developed in the late 1970's by Professor Hiroshi Makino, and has four degrees of freedom (DOF) as shown in Fig. 1.

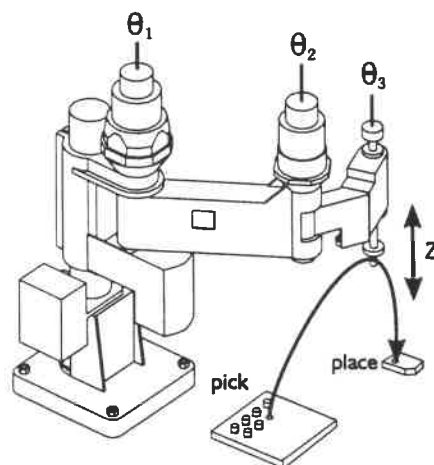


Figure 1: Conventional SCARA assembly robot.

Whereas the SCARA has a proven record and well-known advantages in assembly, it does not lend itself well to the creation of highly integrated systems for precision assembly. A few of the difficulties are:

- large motion range required to access parts feeders
- heavy robot arm must be used to handle very light precision parts
- high accelerations are needed for pick-and-place cycle to sustain high throughput
- serial kinematic linkage with relatively flexible joints can lead to structural oscillation
- dynamic interactions between θ_1 and θ_2 joints must be cancelled by controller
- proximally-located joint encoders for θ_1 and θ_2 limit motion resolution at the last link

Because of these considerations, typical SCARAs used in assembly have motion resolutions and repeatabilities of 50 to 100 μm at best which severely limits their usefulness in high-precision work.

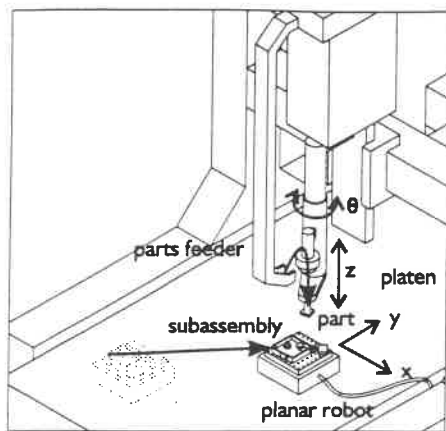


Figure 2: Pair of 2-DOF manipulators used in Mini-factory

2 Modular Robotic Elements

We are developing new modular robotic elements which are key to our Agile Assembly Architecture concept.

Cooperating 2-DOF robots

We propose an alternative robot configuration which provides the same four degrees of freedom as the SCARA but greatly ameliorates these problems. The new configuration is shown in Fig. 2, where the four-DOF SCARA is replaced by a pair of simpler 2-DOF robots.

The θ_3 and z axes of the SCARA are retained and fixed in the workspace. These degrees of freedom can be made highly precise by co-location of sensors and actuators that can be rigidly referenced to mechanical ground. The θ actuator can be carried by the z actuator (or vice versa), or a two-dimensional cylindrical motor design with a single moving part can be used, e.g. [1]. We refer to this robot and its associated feeder as a "manipulator/feeder."

The problematic θ_1 and θ_2 axes of the SCARA are discarded in favor of an x, y stage "courier" robot which carries the subassembly. Moreover, this robot is implemented as a two-axis linear motor capable of traveling above a flat platen surface over a large workspace. This robot is also tightly referenced to mechanical ground. Planar motor robotic systems exist commercially, e.g. [2], but these systems essentially imitate the SCARA configuration.

As indicated in Fig. 2, the two robots must execute coordinated motions in order to replace the functionality of the SCARA. Since the θ, z robot manipulator picks a part which might be only loosely oriented in the feeder, it must perform a rotation in order to correctly align the part in the subassembly, while the x, y courier robot must move the subassembly to the correct position in the workspace. Thus the SCARA uses all of its degrees of freedom for both the pick and the place operation, whereas the new configuration uses two degrees of freedom for dealing with the part, and two separate degrees of freedom for dealing with the subassembly to accomplish the same purpose. This means that the motion required to position the subassembly can be

overlapped in time with the motion required to pick the part (as indicated by the two motion arrows in Fig. 2)—something the SCARA cannot do. Another important feature is that each of the 2-DOF robots can be approximately an order of magnitude smaller in size than the typical SCARA for assembling the same-size product. This leads to a large increase in achievable precision. It also allows the shrinking of conventional automated assembly systems from large room size to tabletop size.

Planar motor sensing and control

Whereas high-performance servo-controlled rotary and linear actuators exist in well-developed forms, unfortunately the planar linear motor is available today only in the form of moderate-performance open-loop microsteppers, which prevents the achievement of maximum potential performance. To help ensure against loss of synchrony (missing steps), only two-thirds to three-fourths of the available force margin is typically used, reducing the potential maximum acceleration and velocity. Even so, susceptibility to loss of synchrony remains if large enough unanticipated external forces are acting. Additionally, settling times after moves are longer than desirable and there is no way to reject low-frequency external disturbances. There is only moderate stiffness requiring high power dissipation when holding a position.

These problems can be alleviated by incorporating a position sensor to accurately measure the relative displacements of the planar linear motor robot and its platen at a bandwidth that is high enough to be used for servo control.

The motor sections have fine teeth (typically 0.5 mm wide on a 1.0 mm pitch) and the platen has a two-dimensional array of square teeth of corresponding width and pitch. After chemical machining, the platen surface is planarized using epoxy to form the air-bearing surface. We are investigating ac magnetic, and optical fluorescence sensing techniques [3]. The magnetic sensor has achieved 1 μ m position resolution and is compact and easy to fabricate. The optical fluorescence sensor has the advantage of complete insensitivity to nearby motor fields.

We are currently working to integrate these sensing techniques into the planar linear motors and plan to use variable structure control methods for greatly enhanced performance.

3 Miniature factories

We are developing prototype miniature factories (minifactories) for precision assembly that are based on the modular robotic elements presented in the previous sections. Platens for the closed-loop planar motors will be in the form of field-joinable tiles, enabling many different assembly line topologies to be realized. Each platen tile is supported by a modular base unit supplying power, air, vacuum, and network services. Modular structural bridges attached to the base units and spanning the platens will support the manipulator/feeders. Additional overhead equipment such as glue-dispensers, lasers,

screw driving heads, and metal forming (spinning, swaging, etc.) tools can be incorporated.

Each active modular robotic element has its own powerful local computing resource and is represented by a software agent. Each agent implements servo control algorithms, maintains state information, communicates with any or all other agents, and contains internal functional and geometric descriptions of itself for use during minifactory modeling, simulation, and programming.

4 Agile Assembly Architecture

Figure 3 illustrates the main features of our proposed AAA. In Fig. 3(a), a manufacturing engineer (minifactory *designer*) is shown interacting with a comprehensive modeling and simulation environment. Functional models of the minifactory components are distributed—they reside within each minifactory module at the geographic location of the module vendor company. When a minifactory model is generated in, e.g., Pittsburgh, the various robotic modules which make up the model might be physically located in Akron, or Detroit, etc. Thus the modeler/simulator in Pittsburgh needs no prior knowledge of the components. Further, by accessing and incorporating data from each on-line module remotely (by Internet), the modeler/simulator has only up to date and reliable information—instead of catalog or faxed data sheet information which might not reflect the currently available module product. During the creation of the minifactory design (model), the designer is free to select and try out (simulate) a wide variety of configurations using modules from many different sources to arrive at a final design (virtual minifactory) shown in Fig. 3(b). The designer may also make use of various tools such as automated assembly sequence planners, schedulers, and layout tools.

From the perspective of the minifactory module supplier or *vendor*, manufacturing modules complying with the minifactory paradigm are designed, built, and programmed by traditional means. Modules offered for use are placed “on the shelf” and plugged into the Internet, thereby becoming available to any minifactory designer. Synchronizing and lockout means are provided to mediate simultaneous engagement by multiple designers. Modules selected by (geographically distant) designers for use in real minifactories are leased according to usage schedules or other criteria and air-shipped to their destinations [Fig. 3(c)] ready for set up and work within hours of their arrival.

From the perspective of the minifactory *builder*, who may be geographically separated from both the designer and module vendors, the minifactory is rapidly set up manually by “snapping together” modules and hooking them up to four based services (power, air, vacuum, network) as illustrated in Fig. 3(d). This is done by following a plan generated as a by-product of the minifactory design process. It is neither necessary to configure the minifactory exactly according to the model, nor to precisely place each of the modular robotic elements. One or more high-performance general-purpose workstations are

connected for modeling, simulation, programming, debugging, and operation monitoring.

Once the (real) minifactory is physically configured and set up, an automated process is begun whereby the subproduct courier modules “explore” their local domains to determine the precise locations and identities of all other modular elements as well as the topology and boundaries of the available workspace. This information is broadcast by each courier module and collected by the workstation modeler. Combined with geometric and functional information from each of the other modules, a complete minifactory model is automatically generated by the minifactory itself [Fig. 3(e)]. This model resembles the original design model, except the new model accurately reflects the actual “as built” minifactory.

From the perspective of the minifactory *programmer*, once the automatically-generated model is complete, it is programmed manually from an attached workstation using mostly graphical methods as illustrated in Fig. 3(f). Each active minifactory module embodies an agent representing it which presents multiple views to the programming environment. For example, the functional controls of a module are accessed and manipulated through its *dashboard* view. A module’s public input and output ports are available through its *interface* view, etc. The minifactory is programmed by graphically connecting outputs to inputs and specifying actions to be performed from internal repertoires. For example, a part handled by a manipulator/feeder module can be aligned with and placed on a sub-assembly carried by a courier module (as in Fig. 2) by specifying a set of coordinating actions which must occur between modules to carry out this task.

From the perspective of the minifactory *operator*, once the minifactory is programmed, execution takes place in an event-driven, asynchronous manner, without supervisory control at any level [Fig. 3(g)]. Attached workstations perform monitoring and debugging functions (in event of non-recoverable failures). Supplying the minifactory with parts on demand is done manually, and removal of finished products is done by other automated or semi-automated means.

At any later time, the minifactory can be reconfigured to respond to changing market requirements [Fig. 3(h)]. Modules can be added or removed, and new minifactory models can be automatically regenerated. At any time, modules no longer needed can be returned to the on-line pool for use by other manufacturers [Fig. 3(i)].

5 Project Status

We began working on several aspects of the AAA in early 1994.

Solid models of minifactory elements and various minifactory layouts have been developed, allowing us to generate animated sequences which include parts placement on subassemblies carried by couriers.

We have invented two position sensing technologies for the planar linear motors that comprise the minifactory couriers [3]. Critical electromechanical

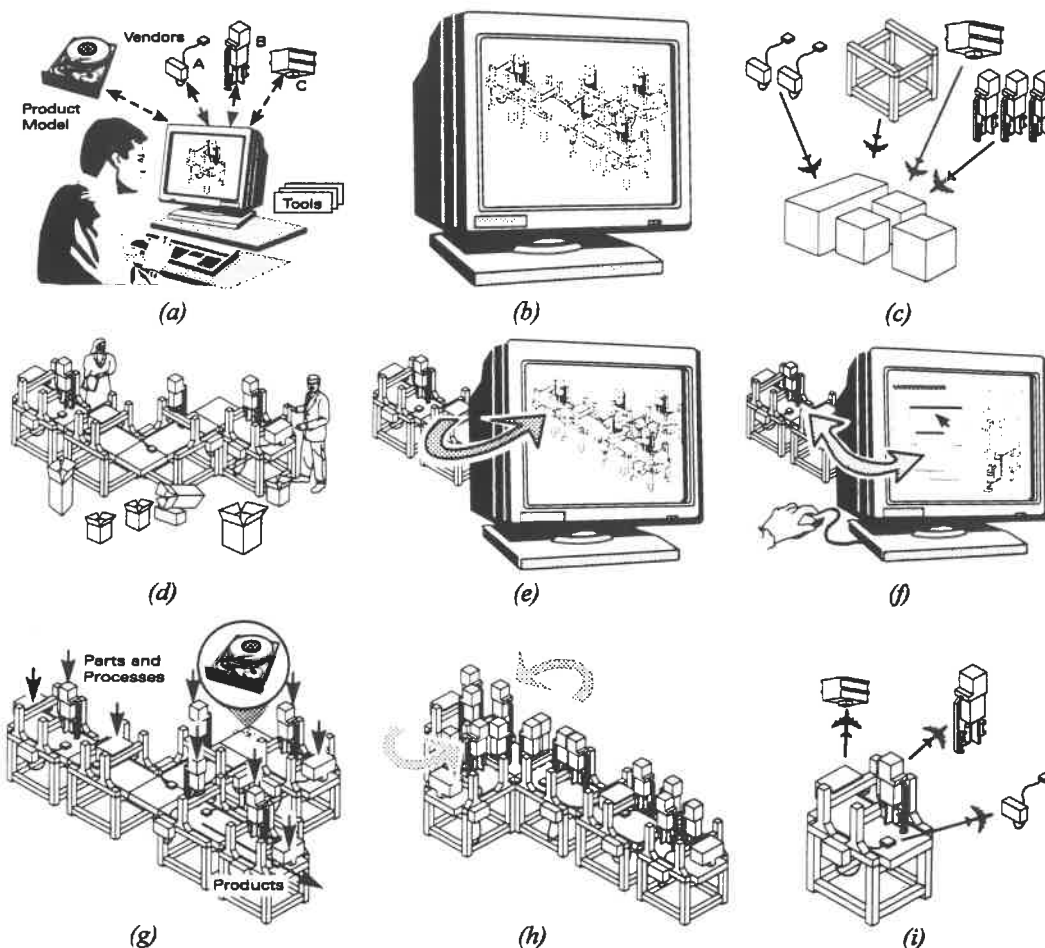


Figure 3: Proposed Agile Assembly Architecture Scenario. (a) Design with access of remote modules via Internet, (b) completed geometric and functional model, (c) ordering and shipment of modules to manufacturing site, (d) rapid assembly of minifactory, (e) automatic generation of updated model from real minifactory, (f) interactive graphical programming, (g) operation, (h) rapid re-configuration to meet changing requirements, (i) re-use of components no longer needed.

properties requiring modeling for the planar linear motors have been identified and we have begun to characterize the key properties. A simulation using a simple motor model demonstrated the dramatic performance improvements possible by adding sensors and closed-loop control.

We have studied the problem of flexible bulk parts feeding in the minifactory context, and have invented a novel singulation mechanism using flexible belts to extract parts from the storage bin, deliver them to the orientation subsystem, and return unselected parts to the bin.

A "Version 0" minifactory based on commercial parts is under construction and partially operating.

Acknowledgements

Many people contributed to the ideas presented here. In particular, we acknowledge the contributions of Patrick Muir for helpful discussions, and Jeroen Vos, Jim Hemmerle, John Kim, and Wen-Cheng Lin for generating 3D animated sequences of

minifactories in action.

This work is supported in part by NSF grants DMI-9523156 and CDA-9503992, and by the CMU Engineering Design Research Center. The Delft University KIVI and STIR programs funded a visiting scholar (Jeroen Vos) for a six-month stay.

References

- [1] R. Hammer, "Combined linear-rotary direct drive step motor." U.S. Patent 5,093,596, March 1992.
- [2] V. Scheinman, "RobotWorld: a multiple robot vision guided assembly system," in *Robotics Research, the Fourth International Symposium* (R. Bolles and B. Roth, eds.), (Santa Cruz, CA), pp. 23-27, MIT Press, 1987.
- [3] A. E. Brennemann and R. L. Hollis, "Magnetic and optical-fluorescence position sensing for planar linear motors," in *Proc. Int'l Conf. on Intelligent Robots and Systems*, vol. 3, (Pittsburgh), pp. 101-107, August 1995.

Breakthroughs in 3-D Ultra-fine Shape fabrication by Dynamic FAB Etching

Masayuki Nakao*, Shuji Tanaka*, Yotaro Hatamura*,

Masahiro Hatakeyama** and Katsunori Ichiki**

*Department of Engineering Synthesis, Faculty of Engineering, The University of Tokyo
7-3-1 Hongo, Bunkyo-ku, Tokyo 113, Japan

**Ebara Research Co., Ltd.

4-2-1 Hon-Fujisawa, Fujisawa-shi, Kanagawa 251, Japan

1 Introduction

A photo-fabrication technique as a semiconductor integrated circuit technology has fostered the increase of the integration density of a LSI by batch print of a mask pattern. As a result, many electronic devices such as a LSI have been available at low cost. Micro-machining as the application of photo-fabrication also has been studied, and micro-devices such as an electrostatic motor and various sensors have been fabricated.

Thus photo-fabrication technology has been one of the key technologies for a semiconductor integrated circuit as well as micro-machining. Three-dimensionality of fabrication, however, is essential for the realization of micro-devices with many functions and micro-machines with complicated structures. Ultra-fineness of fabrication also becomes very important for the substantiation of a LSI and advanced electronic devices with smaller size and more components.

First, in the second section, we discuss photo-fabrication technology, which is the most widely-used technology in micro-fabrication, from the view point of 3-dimensionality of fabrication to reveal its limitation. And we propose a new fabrication method, "Dynamic FAB Etching", to overcome the limitation. As an example of the 3-D micro fabrication, we introduce the experimental fabrication of a micro Fresnel-lens-like object of 260 μ m in diameter. Next, in the third section, we have a similar discussion in terms of ultra-fineness of fabrication, and introduce the experimental fabrication of ultra-fine stairs with a 30nm step.

2 Three-dimensionality of Fabrication

2.1 Limitation of Photo-fabrication

The basic method of the conventional photo-fabrication is a repetitive use of operations such as coating resist, patterning, etching and removing resist, all of which take place on the surface of a workpiece. In other words, the fabrication basically takes place along the single direction to a workpiece, and results in 2-dimensional machining which is perpendicular to the direction.

It is extremely difficult to uniformly coat resist on a non-planar surface because of spin-coating. Therefore, it is difficult to conduct the photo-fabrication to a non-planar surface such as a curved surface or a rough surface. In this sense, it can be said that the conventional photo-fabrication process is inherently 2-dimensional.

Although some 3-D structures can be fabricated by crystal anisotropic etching or sacrificial layer etching, they are comprised of a stack of planar structures, and have relatively small thickness whose range is

several microns to a few ten microns as opposed to several ten microns to several hundred microns in width. From this point of view, they might be classified as "2.5-dimensional" structures. Resistless photo-fabrication free from a resist coating process also has a serious limitation that a workpiece must be a sensitive material.

It can be clearly seen from the above discussion that it is difficult to fabricate a 3-D structure through the conventional photo-fabrication techniques and that a new method needs to be introduced to circumvent the limitations.

2.2 Dynamic FAB etching

The conventional methods suffer from the above-mentioned limitations in terms of 3-dimensionality of fabrication. In order to overcome the limitations, we propose a new method using a FAB (Fast Atom Beam) which has an outstanding rectilinearity and a non-contact mask separated from a workpiece. A FAB consists of extremely straight flux of fast atoms with a speed of approximately 100km/sec., which are generated through a cathode by electrical neutralization of positive ions which are ionized and accelerated by high voltage of several kV between an anode and a cathode. In addition to its excellent rectilinearity, a FAB can make an etched side wall nearly perpendicular, and can etch non-conductive materials such as glass and polyimide due to its electrical neutrality.

The proposed fabrication method is in principle an etching process under the control of the straight beam flux projected to a workpiece using the coordinated motions of the non-contact mask and the workpiece which are set apart from each other. A FAB etches the workpiece in the same pattern as the one the non-contact mask has despite the distance between the mask and the workpiece. We call this fabrication "Dynamic FAB Etching", where the term "dynamic" comes from the dynamic motions of the non-contact mask and the workpiece.

In this method, multi-directional fabrication is realized because the mask and the workpiece, which are separated from each other, can freely translate and rotate. The fabrication to a non-planar surface can also be realized because of the absence of resist. For the above reasons, the proposed fabrication method proves to be appropriate for 3-D fabrication. Figure 1 shows a couple of examples of the 3-D fabrication by the coordinated motions of a non-contact mask and a workpiece, whose details are explained in the next section.

2.3 Experimental Fabrication of a micro Fresnel-lens-like object

We conducted the experimental fabrication of a mi-

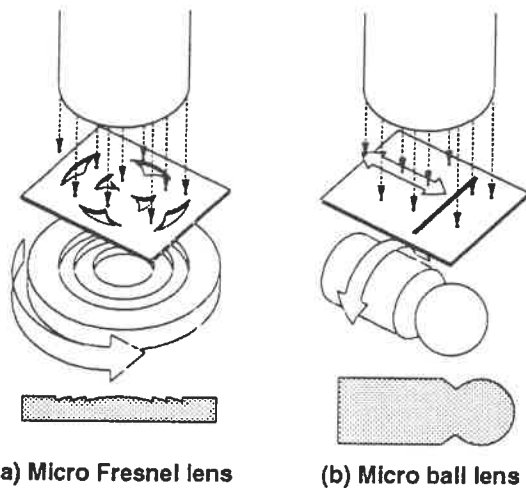


Fig. 1 Process for fabricating 3-D structures

cro Fresnel-lens-like object as an example of the 3-D fabrication by the dynamic FAB etching[1]. Through the experiments, we aim to confirm the feasibility of the fabrication of the micro 3-D structure whose size is the order of $100\mu\text{m}$. Figure 1(a) shows the schematic of the experiment. The workpiece rotates about the vertical axis at a constant speed. In this case, the center angle of the mask hole determines the time interval of FAB projection to the workpiece, to which the etching depth is proportional.

We used Cl_2 as a FAB gas, a GaAs substrate as a workpiece and a Ni electric molding mask $5\mu\text{m}$ thick as a non-contact mask which was fabricated by patterning with photolithography shown in Fig. 2. We developed a rotational mechanism for a workpiece which is equipped with a couple of precise angular contact ball bearings and a motor installed outside a vacuum chamber.

Figure 3 shows the experimental result after 11 hours of etching with the workpiece rotating at a speed of 5 rev./hour after setting the non-contact mask approximately $100\mu\text{m}$ apart from the workpiece. Being judged from the surface profile measured by an probing surface roughness tester shown in Fig. 4, the Fresnel-lens-like object with three zones is successfully fabricated as intended. The etched mark is approximately $260\mu\text{m}$ in diameter and $8.2\mu\text{m}$ in depth, and the texture of the etched surface is smooth. The edges of the slopes are not sharp in the range of approximately $10\mu\text{m}$ including the measuring error due to the probe tip of the probing surface profile tester ($\phi 2\mu\text{m}$). It is caused by various factors such as the imperfect setting of the center of the mask and the rotational center of the workpiece, the misalignment of the rotational axis of the workpiece, and the positional offset of the mask due to the thermal deformation induced by the FAB projection in a vacuum, whose resolution will be the future work.

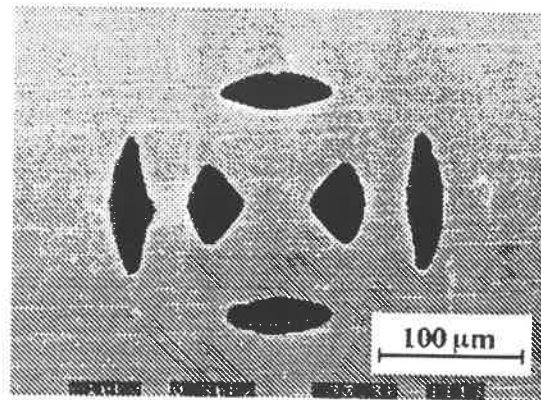


Fig. 2 Non-contact mask for a micro Fresnel-lens-like object

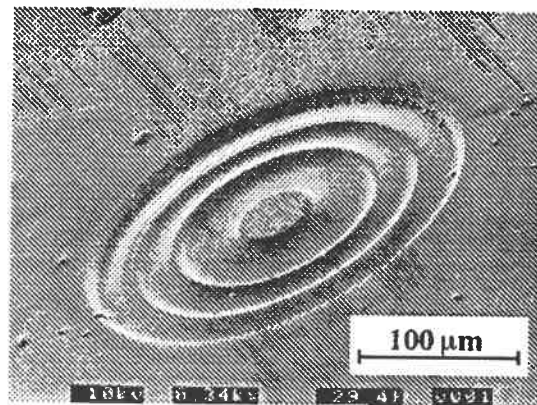


Fig. 3 Micro Fresnel-lens-like object

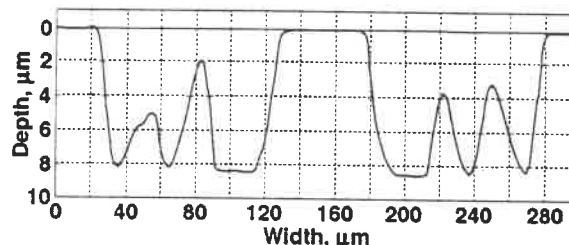


Fig. 4 Surface profile of the micro Fresnel-lens-like object

3 Ultra-fineness of Fabrication

3.1 Limitation of Photo-fabrication

In the conventional photo-fabrication, the precision depends on the precision of a mask pattern. Many researches in improving the precision of a mask pattern have been carried out by the employment of exposure rays with smaller wavelength for printing a mask pattern and the reduction of the spot diameter of an electron beam for mask pattern drawing. There is, however, a limitation that the precision of the fabrication is more than the order of $0.1\mu\text{m}$ due to various factors such as the wavelength of exposure ray, the diameter of electron beam spot, and the thickness of resist. On the other hand, the ultra-fine fabrication with nanometer order accuracy is required in the field of a single electron device, which is difficult to realize by conventional methods.

3.2 Possibility of the ultra-fine fabrication by the dynamic FAB etching

In the dynamic FAB etching process, a complicated mask pattern for the conventional photo-fabrication can be replaced by a non-contact mask with a simple pattern and a programmed precise motion. For example, in the conventional process, the mask pattern with a line of 50nm in width is required for the fabrication of a groove whose width is 50nm . On the other hand, in the new process, grooves of 50nm in width can be fabricated at intervals of 200nm by the motion of the non-contact mask with a slit of 250nm in width, as shown in Fig. 5. In this case, the precision of the fabrication depends on not the precision of a mask pattern but the precision of the mask motion. If the precise mask motion of nanometer order is realized, the ultra-fine fabrication with nanometer order accuracy can be achieved.

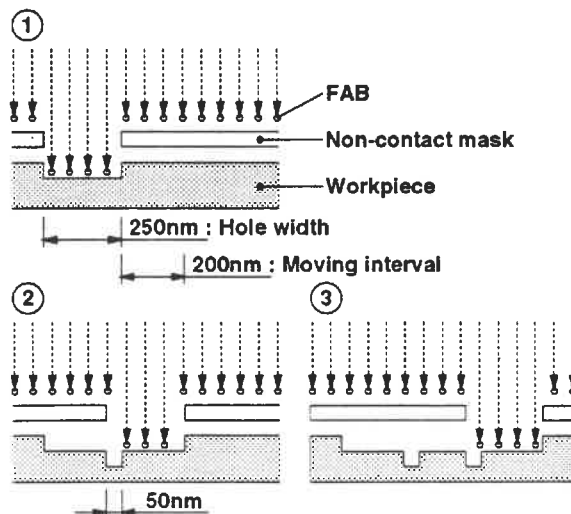


Fig. 5 Process for fabricating grooves of 50nm in width

3.3 Experimental fabrication of a ultra-fine stairs

We conducted another experiment of building ultra-fine stairs as an examples of the ultra-fine fabrication by the dynamic FAB etching. Through the experiments, we aim to confirm the feasibility of the ultra-fine fabrication with 10nm order accuracy. Figure 6 shows the schematic of the experiment.

We again used Cl_2 as a FAB gas, a GaAs substrate as a workpiece and a Ni mask $10\mu\text{m}$ thick or a Si mask $350\mu\text{m}$ thick as a non-contact mask. We developed a precise and coarse piezo actuator for the precise motion of a non-contact mask. The use of a Super Inver alloy as the material of the most part of the actuators significantly reduces the positional shift of the mask due to thermal deformation. In the experiment, we used mainly a couple of the fine piezo actuators connected in series, which is designed to realize the resolution of 1nm and the full working range of $1\mu\text{m}$ by the mechanism for displacement reduction shown in Fig. 7.

The surface profile measured by an AFM (Atomic Force Microscope) shown in Fig. 8 exhibits the experimental result with the Ni mask $10\mu\text{m}$ thick. In this experiment, two times of etching for 2 minutes took place, accompanied with the mask translation parallel to the workpiece from the low ground of the stairs to the high one at an interval of 30nm . The ultra-fine stairs with 2 steps whose width is 30nm and height is approximately 30nm was successfully fabricated as intended, and the ultra-fine fabrication with 10nm accuracy is realized.

The surface profile in Fig. 9 exhibits another experimental result with the Si mask $350\mu\text{m}$ thick. In this experiment, five times of etching for 5 minutes took place, accompanied with four times of the mask translation parallel to the workpiece from the high ground of the stairs to the low one at intervals of $100\sim 500\text{nm}$. The ultra-fine stairs with 5 steps whose width is $100\sim 500\text{nm}$ and height is approximately 100nm was fabricated. The edges of the stairs, however, are blunt in the range of $50\sim 100\text{nm}$ including the measuring error due to an AFM probe tip, which is probably caused by the use of the thick mask.

Being judged from the abovementioned results, the use of a thinner non-contact mask reduces the range of the blunt etched edge. It is probably caused by the fluctuation of FAB projection due to the reflection of a FAB on the side wall of a non-contact mask, as shown in Fig. 10(a). According to other experiments, the range of blunt stair edges tends to increase as the distance between a non-contact mask and a workpiece becomes large. When the distance is approximately $100\mu\text{m}$, the range of a blunt stair edge becomes several hundreds nanometer. It is probably caused by the reflection of a FAB and the deposition of an etched workpiece onto the surface under a non-contact mask where a FAB is not projected, as shown in Fig. 10(b). In order to obtain the objective accuracy of less than 10nm , it seems necessary to precisely move a non-contact mask whose thickness is the order of $1\mu\text{m}$ apart from a workpiece by less than $1\mu\text{m}$. The improvement in this method and the fabrication of ultra-fine grooves by the method shown in Fig. 5 will be the future works.

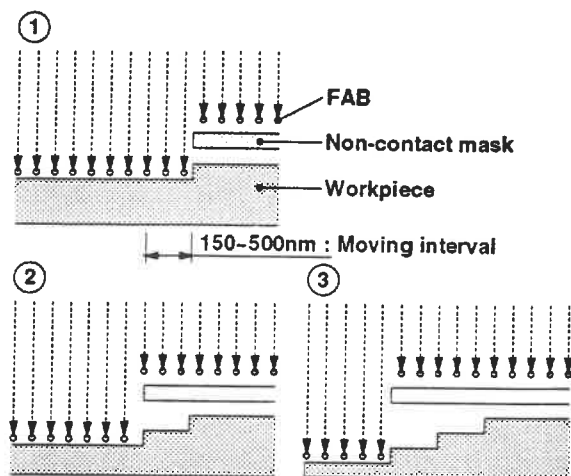


Fig. 6 Process for fabricating ultra-fine stairs

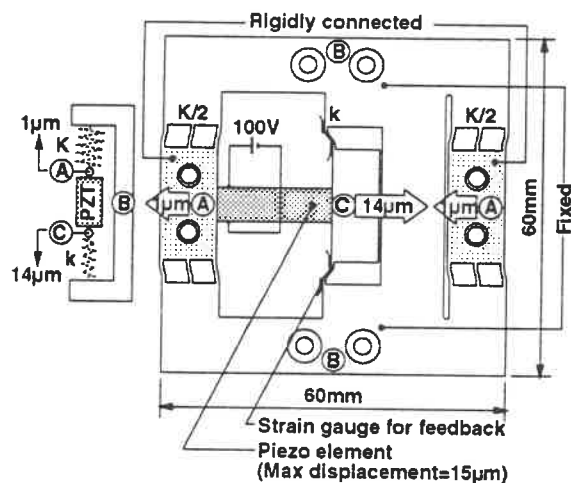


Fig. 7 Schematic view of the fine piezo actuator

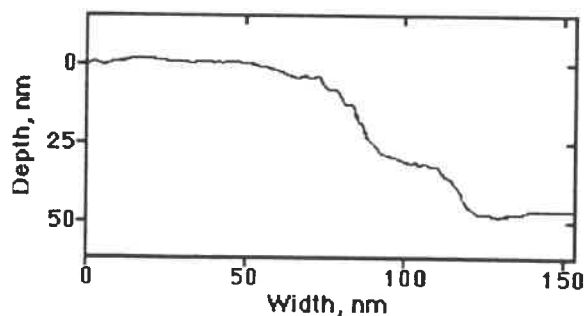


Fig. 8 Cross section of the ultra-fine stairs fabricated with the thin Ni mask



Fig. 9 Cross section of the ultra-fine stairs fabricated with the thick Si mask

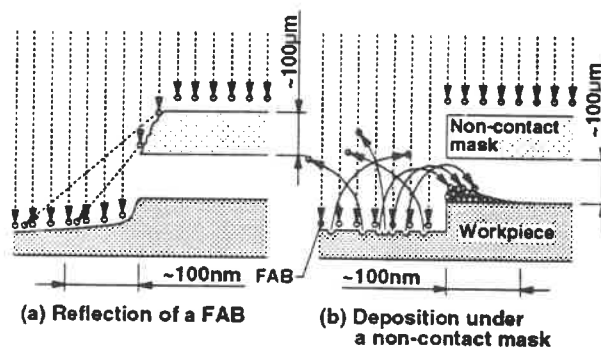


Fig. 10 Micro phenomenon near the non-contact mask

4 Conclusion

We proposed the dynamic FAB etching utilizing a FAB and a non-contact mask, aiming at overcoming the limitations of conventional photo-fabrication in terms of 3-dimensionality and ultra-fineness of fabrication. As regards 3-dimensionality of fabrication, we fabricated a micro Fresnel-lens-like object of 260 μm in diameter, while ultra-fine stairs with a 30 nm step were fabricated to examine the aspect of ultra-fineness. We believe that the dynamic FAB etching will be a good candidate to overcome the problems that conventional methods suffer from and will make a significant contribution to the realization of a useful micro-machine and a single electron device.

Reference

- [1] M. Nakao and S. Tanaka, et al., Experiment of Slope Etching by a Multi-face FAB, Proceedings of JSME 72th Spring Annual Meeting, 1995, Vol. VI, pp. 265-268

DESIGN AND FABRICATION OF A TWO DIMENSIONAL VALVELESS MICROPUMP

W.K. Kahl, C.M. Egert and K.W. Hylton
Oak Ridge National Laboratory
Oak Ridge, TN 37831-8039

Introduction

The scale-down of a liquid mini-pump (order of 10 mm) to a micrometre scale has been attempted using a novel valveless nozzle-diffuser design and new application of an organic physical vapor-deposited membrane [1,2]. The micropump employs no moving parts other than the membrane and accomplishes the rectification of fluid flow due to pressure recovery differences in the nozzle and diffuser flow directions. More specifically, liquids flow with less resistance (i.e. conduct more fluid) in the diffuser direction than the nozzle direction, for a given pressure differential. At the micrometre scale, the fabrication of the critical nozzle and diffuser elements was performed by focused ion beam (FIB) microlithography of glass slides. Etched slides were sandwiched to make two-dimensional venturis. Stemme and Stemme noted the importance of a lower Reynolds Number limit on the desired pressure recovery which challenged the fabrication of this pump design at the scale used.

Pump Design

The pump makes use of a fluidic diode, or device which has no moving parts and acts as a check valve by imposing low resistance to flow in one direction and high resistance when the flow is reversed [3]. The disadvantage of power fluidics such as this design are (1) the diode adds resistance to flow in the forward direction and (2) it will not stop backflow completely (Fig.1). The net result is loss of volumetric pumping efficiency as the trade for mechanical reliability.

The pumping mechanism is accomplished by simple suction and discharge strokes. In the suction stroke, fluid is drawn through both venturis, with the nozzle converting static pressure to dynamic pressure as the fluid accelerates through the throat. The diffuser section recovers the static pressure (minus losses) by decelerating the flow in an orderly fashion, not in chaotic eddies which dissipate kinetic energy as internal energy.

The design of the micropump was based upon a two-dimensional approximation of the three-dimensional conical diffuser / nozzle geometries utilized by Stemme and Stemme [1,4]. This approximation was performed by etching nozzle / diffuser shapes into 100 μ m thick glass cover slides to create planar (i.e. rectangular cross-section) channels.

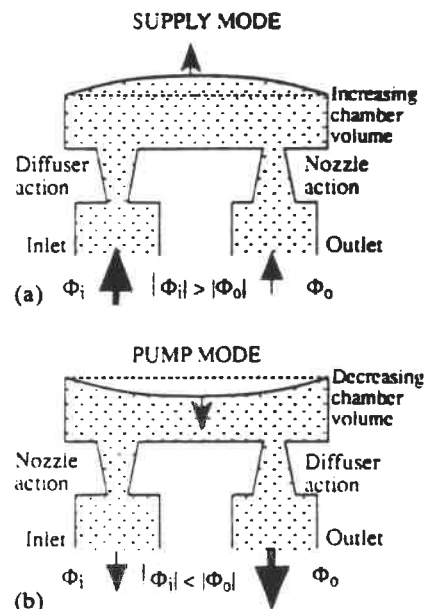


Figure 1. Pump operation principle
(Φ is volumetric flowrate)

Fabrication Process

For demonstration of principle and ease of construction at the microscale, a two dimensional venturi was constructed as opposed to an axisymmetric model. The fabrication steps for the micropump included thin film metallization of the glass pump body and cover slides, focused ion beam machining (FIB), wet chemical etching and microassembly processes. The FIB was an FEI Model 200 System utilizing a gallium beam at 10,000 pA. The beam profile spot size was 600nm wide (which is related to the the full width of the beam at half the maximum current), necessitating a lengthy and repetitive pattern coverage.

The initial step involved the creation of the nozzle / diffuser pattern and plenums by rastering a FIB gun over a chromium and gold-plated glass slide to remove the plate. Gold was used because the hydrofluoric acid (HF) solution will not etch gold, but the latter does not adhere well to glass. Chromium was used as an intermediate layer to address this. The 100Å thick chromium plating was sputter deposited on the glass, and the 1000Å coating of Au was applied onto the Cr. After ion milling the desired patterns through the metal film, the slide was then chemically etched with a dilute solution of hydrofluoric acid to create the hydraulic channel in the glass. The acid etch depth here was 11 µm and essentially unidirectional.

The micropump as designed is 850 µm by 600 µm in footprint. The etched pump layout is shown in Figure 2. The pump chamber plenum is 300 µm long and 600 µm wide. The nozzles are each 250 µm wide at the mouth and contract to 50 µm wide in the throat. Diffuser sections are 50 µm wide in the throat and expand to 100 µm wide at discharge. Diffuser lengths are 300 µm. The flow suction plenum is 350 µm by 150 µm and the flow discharge plenum is 200 µm by 150 µm.

The cover slide was 100 µm thick glass and was modified on one side with a 2.5 µm thick membrane of parylene adhesively attached. The reverse side was then gold coated and patterned similarly to the pump volume with cavities registered over the pumping plenum and suction / discharge plenums. This side was etched completely through the glass with a mild HF solution until the parylene was exposed. Registration of the cover slide over the plenums provided a flexible membrane for pumping action. This membrane was actuated by a Burleigh PZ piezoelectric pusher with a 10 µm stroke. The actuator amp was driven by a Hewlett-Packard waveform generator employing a 300Hz sinewave, although other frequencies were tested. The pusher face had a common straightpin affixed with the tip blunted to a 40µm diameter for contact with the parylene and transfer of stroke motion. In addition, 105 µm diameter holes were etched through the cover slide for flow capillaries into and out of the suction / discharge plenums.

Flow Analysis

The basic relations used in this analysis are the energy conservation equation and the continuity equation. Assuming incompressible flow, with no introduction of shaft work, the energy conservation equation reduces to a simplified Bernoulli's flow equation:

$$[P_1/\rho g] + [V_1^2/2g] + [z_1] = [P_2/\rho g] + [V_2^2/2g] + [z_2] + \text{losses}$$

Bernoulli's equation contains three identifiable elements contributing to the total pressure: a static "pressure head" term (P), a dynamic "velocity head" term ($V^2/2g$) and an elevation term (z) which can be ignored for flow analysis in the same horizontal plane. It is immediately obvious that as fluid accelerates through a nozzle throat or decelerates through a diffuser, there is a transfer in the energy contained in these terms and viscous losses must be incurred which generally manifest as pressure drops and heat lost via turbulence, etc. Such losses are characteristically modeled with empirical coefficients applied to maximum

values of the dynamic term for the region of interest. Addition of these loss expressions to the Bernoulli equation, and substitution of the continuity expressions (channel cross sectional area x velocity),

$$A_1 V_1 = A_2 V_2$$

can be combined to arrive at simplified equations for flowrates, downstream pressures, etc. as functions of known values and loss coefficients. Derivation of these expressions is beyond the scope of this paper and can be found in Ref. [1]. Importantly though, pump flowrates can be modeled as functions of the pump chamber volume oscillation modified by a factor accounting for the geometry-specific loss coefficients.

The calculation of an appropriate laminar / turbulent transition Reynolds number for this micropump was suggested by Gravesen et al.[2] as,

$$Re_t \approx 30(L/D_h)$$

where $L=300\mu\text{m}$ as the diffuser length (assumed equal to entrance length for fully-developed flow) and D_h is the hydraulic diameter obtained by four times the channel cross-sectional area divided by the wetted perimeter. The hydraulic diameter was computed for the $50\mu\text{m}$ by $11\mu\text{m}$ throat as $18\mu\text{m}$ yielding an L/D_h ratio of 16.6, and a $Re_t = 500$ (indicative of a short channel restriction). By treating this as an upper limit, a maximum permissible throat velocity of 2.8 m/sec was calculated. Thus, it is evident that all flow was likely in a laminar flow regime. The importance of this implication is that the flow rectification may indeed be occurring in both turbulent and laminar regimes, where only the former was expected. This would suggest successful scale-down of the original mini-pump design.

Pump Performance Tests

Performance data were not available for publication at press time for this paper. Results will be discussed in the meeting presentation. At the macroscale, flow separation along the walls of the diffuser can occur in adverse pressure gradients (i.e. increasing pressure in the direction of flow). For unfavorable gradients, boundary layers can grow and break away from the walls, leading to separation on one side only (bistable steady stall), oscillation from one side to the other (bistable unsteady stall), or occur along both sides (jet flow). Each circumstance results in very poor performance. However, before bistable steady stall develops, transitory stall occurs and is accompanied by highly unsteady flow. Maximum values of pressure recovery have been observed in this region. A stability map of diffuser flow patterns (Fig. 3) provides insight into the tenuous nature of diffuser behavior, and sensitivity to variations in geometry [3,5]. The particular pump design used here had a diffuser angle (2Θ) of 11° and a length-to-throat ratio of 6:1. On Figure 3, this design point places the pump very near the $C_{p\text{MAX}}$ line.

Conclusions

The fluidic analysis of microdevices such as this make use of critical assumptions which bear examination. The assumption that viscosity is independent of the hydraulic dimensions may be invalid at this scale. The polar nature of common pumped liquids (water, oil and alcohol) may have a significant effect [2]. Further more, the debilitating effect of small gas or air bubbles in microchannels can prevent proper pump-priming. The heightened importance of surface tension effects at the microscale is suspect and not well-treated in the literature, either in experiment or theory.

Proposed uses of such a microfluidic pump as this include medical / drug metering, chemical analysis, and microfluidic amplifiers "on-a-chip". If fabricable in batches, this micropump concept could be extended or "ganged" to more macroscopic applications.

Acknowledgements

This work was sponsored by an internally-funded program in Microfabrication at ORNL. Oak Ridge National Laboratory is managed for the Department of Energy by Lockheed Martin Energy Systems, Inc. under contract DE-AC05-84OR21400.

References

- [1] Stemme E. and Stemme G., *A Valveless Diffuser / Nozzle-based Fluid Pump*, "Sensors and Actuators A.", Vol. 39 (1993), pp. 159-167.
- [2] Gravesen P., Branebjerg J. and Jensen O.S., *Microfluidics - A Review*, "IEEE Transactions- ", pp168-182.
- [3] Counce R. and Smith G., *Fluidic Pump Development*, ORNL/TM-7519, Oak Ridge National Laboratory, Oak Ridge, TN., Dec. 1980.
- [4] Kahl W., *Design and Analysis of a Venturi-like Reverse Flow Diverter*, M.S. Thesis problem, Univ. of Tennessee, 1984, unpublished.
- [5] Rundstadler P., Dolan F. and Dean R., *Diffuser Data Book*, TN-186, Create Technical Information Service, Hanover, N.J., 1975.
- [6] Adams R., *Application of Fluidic Valves*, American Soc. Mech. Engr.s Tech. Paper 992, presented at the 1965 Winter Annual Meeting, Chicago, Ill. Nov. 1965.

Keywords: Microfluidics, micropump, valveless, diffuser / nozzle

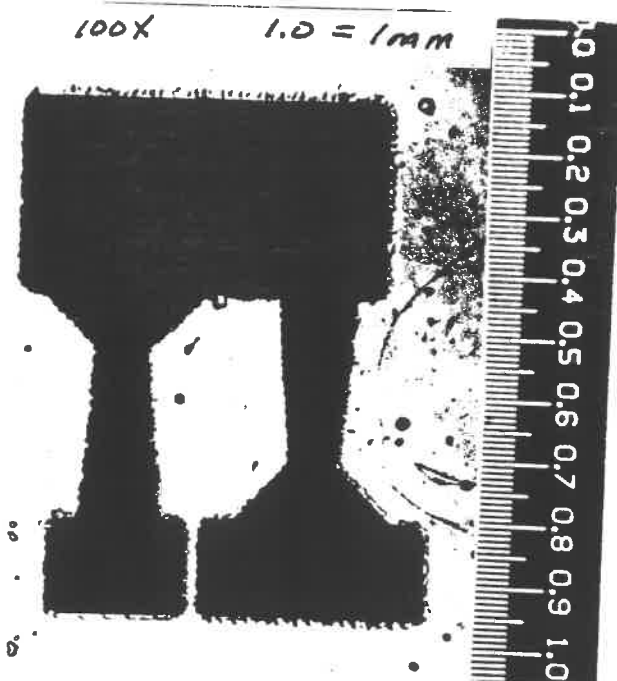


Figure 2: Optical photograph of micropump etch pattern.

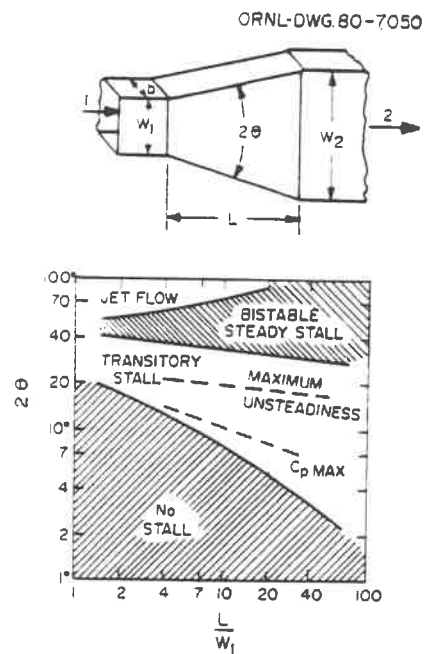


Figure3: Stability map for diffuser designs at all scales.

Micro-Pattern Control by Miniature Robots in Vapor Deposition Process

Hisayuki Aoyama, Kenichiro Mizutani, Futoshi Iwata and Akira Sasaki

*Division of Intelligent Instrument and Information Integration
Department of Mechanical Engineering
Shizuoka University
3-5-1, Johoku, Hamamatsu, Shizuoka 432, Japan*

1. Introduction

For these ten years, it is well-known that several remarkable machine tools and techniques have been developed for ultra precise machining process of bulk materials in the area of nanometer. Also non-mechanical machining methods such as photolithography for IC process have been established for microfabrication of material deposition and removal. In general, those facilities have to involve the ultra precision positioning mechanism which is incorporated with sensory elements and actuation devices in order to improve such positioning accuracy. A lot of sophisticated efforts which are based on the *top-down approach* have been given to those mechanical systems.

Recently our group have alternatively developed many miniaturized robots with precise functionality such as single point diamond micro machining[1], surface topography monitoring[2], micro hammer indenting[3] and capillary capturing probe[4]. Particularly small and simple mechanisms can provide a very high mechanical stability which is useful for even microscopic operation and should give the great cost-saving benefits. Our goal is to construct precise manufacturing system organized by miniature robots[5] from the view point of *bottom-up approach*.

Another advantage of our small and simple robots is that they can be easily dispatched to the special environment such as high vacuum condition where the performance of normal mechanism become serious due to lubrication problem. So we attempt to design a small robot to work in high vacuum condition and to control the vapor deposition process there. With respect to micro surface modification, it should be valuable that several small robots with micro mechanical tools could collaborate with another robot capable of controlling the local deposition process within a single vacuum operation although conventional facilities such as turning machine never coexists with physical deposition process as shown in Figure 1.

This report describes the simple idea of micro pattern control by miniature robot in vacuum chamber and results of preliminary experiments for the feasibility. The experiments reported here concentrate on making several micro patterns of material coating on a glass sample by precise movable template on miniature robot.

2. Deposition Control by Miniature Robot

Figure 2 shows the schematics of experimental set up. There is a platform on which small robot can walk in a vacuum chamber. This platform has a center hole on where several thin templates are attached so that vaporized metal can go through from the source. Metal

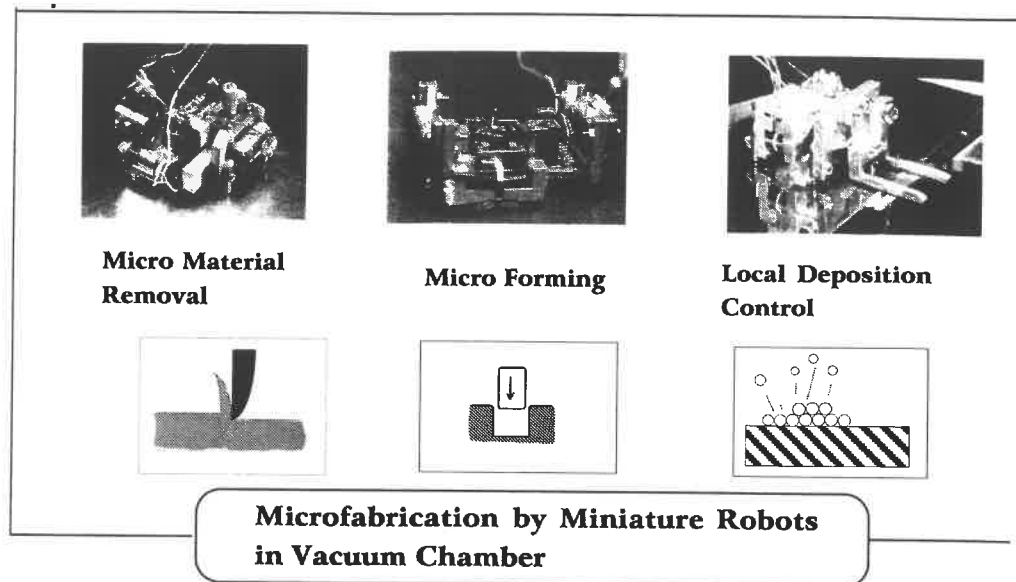


Fig.1 Combination of miniature robots with micro tool in vacuum chamber

source which is heated electrically is set at the bottom and a pin hole plate is located between the source and the template in order to produce narrow distribution.

Figure 3 illustrates small robot with a sample holder. Basically it consists of piezo elements for propelling and electromagnet for clamping. This arrangement can provide precise step motion in any direction without well-finished guideway. Furthermore several improvements such as ceramics coating to solenoid, special adhesive, polytetrafluoroethylene thin wire which are famous in vacuum technology have been made against baking preparation, vacuum condition of even 10^{-5} torr and metal deposition. This small robot can take the sample and position it on the thin

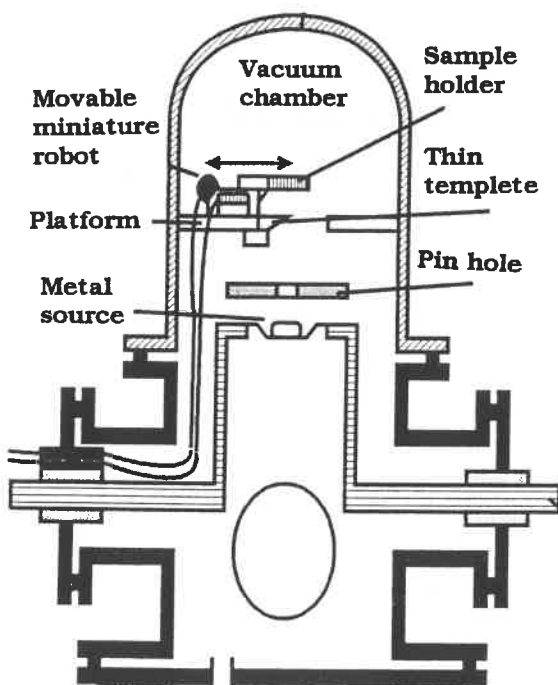


Fig.2 Experimental setup

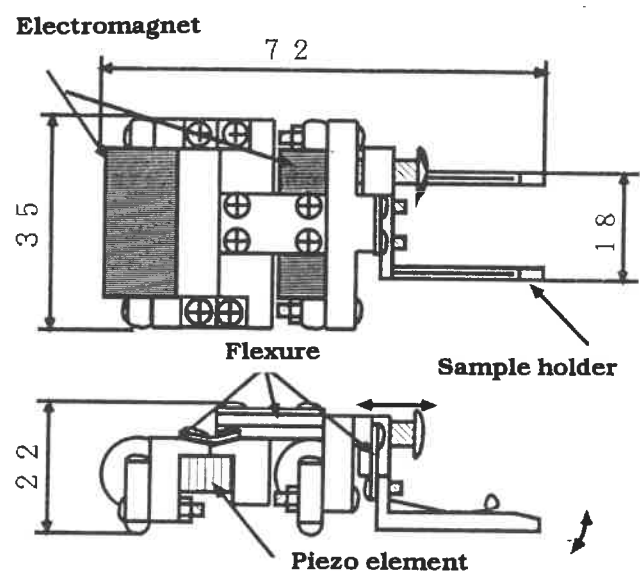


Fig.3 Miniature robot with sample holder

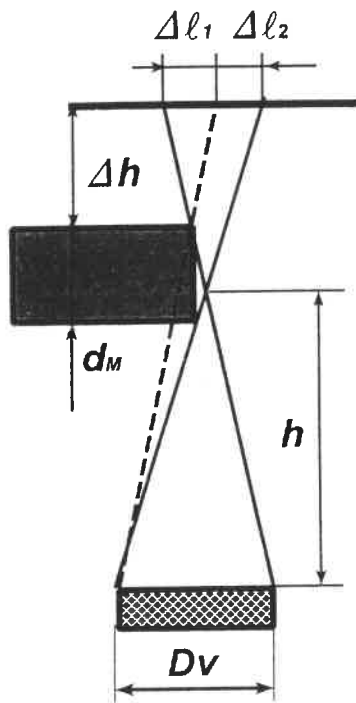


Fig.4 Layout of sample, template and source

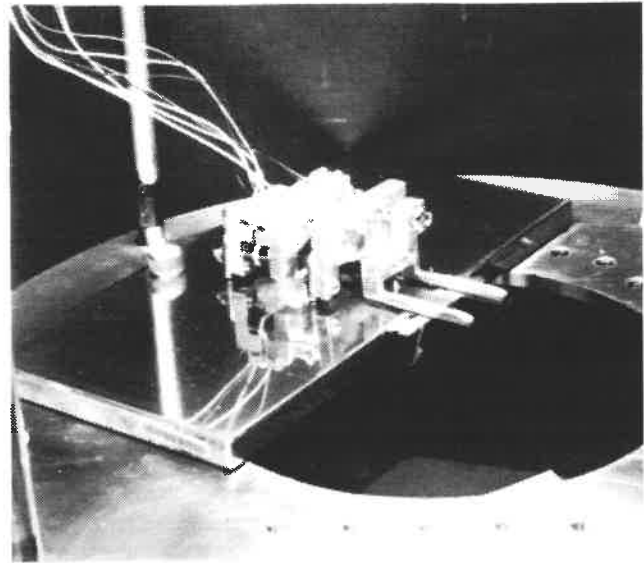


Fig.5 Photograph of miniature robot on platform

template precisely in vacuum chamber. After the first deposition, he can move the sample to next position and then heat up the source again for the second deposition. This simple sequence can generate micro patterns based on the feature of template easily while thickness of thin pattern is

depend on the evaporating time.

Figure 4 shows the geometrical layout of sample, template and material source in chamber. Here uncertainty of deposition edge can be considered simply as follow.

$$\Delta l_1 = \frac{\Delta h}{h} D_v, \quad \Delta l_2 = \frac{d_M}{2h} D_v, \quad \Delta l = \Delta l_1 + \Delta l_2$$

where uncertainty caused by thickness of template(d_M) is Δl_1 , that of source diameter(D_v) is Δl_2 , and h is the distance between the source and the template, Δh is gap between the template and sample. In case of $d_M = 0.01\text{mm}$, $D_v = 2.0\text{mm}$, $h = 200\text{mm}$ and $\Delta h = 0.1\text{mm}$, then total uncertainty Δl of $1\mu\text{m}$ can be estimated.

3. Experimental Setup and Results

Figure 5 is the photograph which indicates that the small robot with sample holder is working on the platform for deposition operating in vacuum chamber. He has a glass plate of $18\text{mm} \times 18\text{mm}$ on his head. Also he is remote-controlled to take it on several templates such as slits and holes. Vaporizing time is switched manually by an operator to obtain appropriate film thickness. Figure 6 is the optical microscope photograph which shows typical result of micro pattern of aluminum which is composed of circle and line when he takes the sample on hole and thin slit. Another result is demonstrated in Figure 7 where he position the sample with $100\mu\text{m}$ spacing on wide slit and hold it again in rotation of

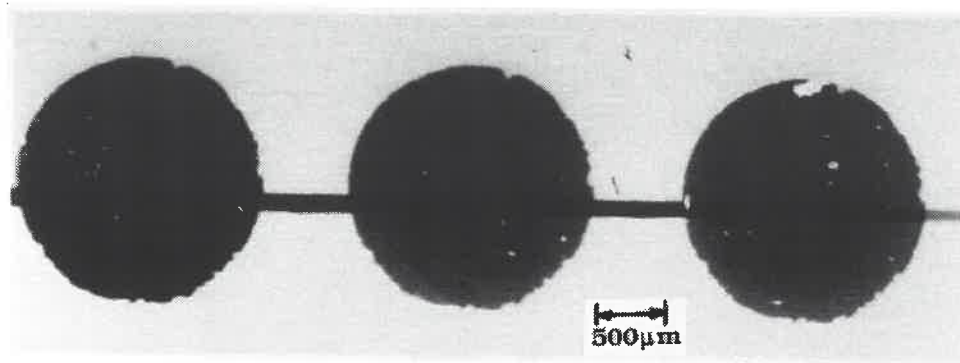


Fig.6 Controlled pattern of three circles and thin line

90 degrees. He succeeded in producing micro checker pattern of two layers made of Al.

4. Conclusions and Future Works

We reported that we made miniature robot and simple experimental set up for controlling local deposition process. This small robot was improved to work in such vacuum condition by using special parts. Experimental demonstrated here were that it succeeded in making several micro patterns of deposition which were made of aluminum on a glass sample while edge quality and thickness have not be measured. So we should evaluate precisely the quality of deposition pattern with help of such as scanning probe microscope. Also we endeavor to make the combination system of micro mechanical machining, micro forming and micro patterning which are performed by miniature robots in vacuum chamber, and to improve it for achieving local reactive process such as sputtering. Parts of this research were sponsored by Tateisi Science and Technology Fundnation.

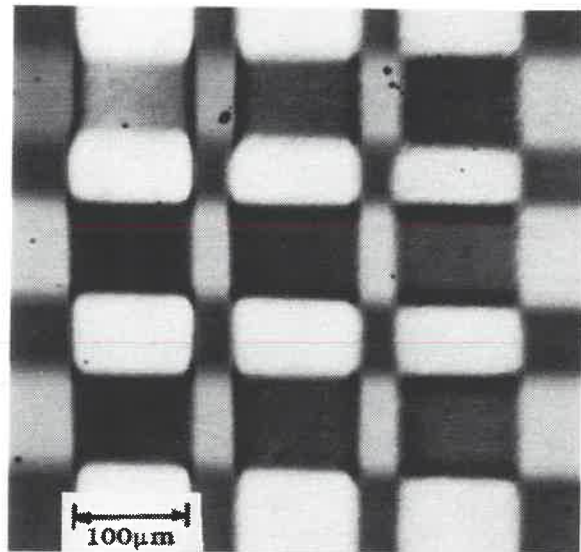


Fig.7 Two layers of checker pattern of 100 micro width

Reference

- [1]H.Aoyama etal; Collaboration Precise Operation by Two Miniature Robots-Single Diamond Cutting with Local Position Feedback, Proc. of 9th Annl. Meeting ofASPE(1994)pp.238-241
- [2]H.Aoyama etal; Precision Machining and Measurement Organized by Miniature Robots, Proc. of 8th Annl. Meeting of ASPE(1993)pp.126-129
- [3]T.Kato etal; Micro Hammer Forming Performed by Miniature Robot, Proc. of Int. Conf. on Prec. Engg. (Nov. 1995) to be printed.
- [4]H.Aoyama etal; Miniature Robot with Micro Capillary Capturing Probe, Proc. of 6th Int. Symp. on Micro Machine & Human Science(Oct. 1995) to be printed
- [5]H.Aoyama etal; Desktop Flexible Manufacturing System by Movable Miniature Robots-Miniature Robots with Micro Tool and Sensor, Proc. of IEEE Int. Conf. on Robotics & Automation, Vol.1(1995)pp.660-665

NEW ALGORITHMS FOR THE EVALUATION OF DISCRETE POINT MEASUREMENT DATA AND FOR SAMPLE POINT SELECTION ON SURFACES WITH SYSTEMATIC FORM DEVIATIONS*

K. D. Summerhays, R. P. Henke, R. M. Cassou, University of San Francisco, San Francisco, CA; J. M. Baldwin, Sandia National Laboratories, Livermore, CA; and C. W. Brown, Allied Signal Aerospace, Kansas City, MO

ABSTRACT

The use of coordinate measuring machines (CMMs) and other discrete point sampling devices in dimensional metrology has raised questions regarding the proper interpretation of data which technically constrains a surface only at the points measured. Reconciling the intrinsic limitations of such inspection techniques with dimensioning and tolerancing standards which often make the implicit assumption that surfaces can be fully characterized has been an issue of major concern in the metrology community in recent years. In addition economic considerations argue for more efficient and reliable procedures for dimensional inspection. This has led to the desire to find sampling and data analysis methods by which the information available through discrete point sampling is maximized.

The work reported here is based on recognition that, while a certain portion of form deviation is unquestionably best treated as random, usually there is also a component which should be treated as systematic. Yet, the most commonly encountered data analysis algorithms in current software systems of CMMs and related devices are based on the notion that *all* of the deviation is random. In most instances the sample point data is fitted by a least-squares algorithm to an ideal geometric surface. This fitted surface then serves as the substitute geometry for the feature of interest and some measure of the residuals of the fit is taken as a gage of the form parameter. For example, the range of the residuals about a fitted plane may serve as an estimate of the flatness for a nominally planar surface. The least-squares method has been favored both because of its computational simplicity and also because it makes statistical use of all the data points in what is ordinarily a very sparse sampling of the surface.

Much less straight-forward computationally, and far less stable statistically, are the minimum zone methods. Here the substitute geometry is established on the basis of a few contact points. The other points sampled serve only to define which are the bounding points. If the surfaces of interest were fully characterized, then these methods would in many instances best embody the spirit of the gaging standards. The use of these methods in standard inspection has been relatively uncommon. This seems appropriate, given the sparse nature of the typical sampling. The objective of our research has been to determine to what extent the recognition of systematic form deviations can yield efficiencies in inspection.

The thrust of our approach is to establish, through sparse sampling, an approximate representation of the real surface over its entire domain. This representation begins with a model surface which captures, as well as possible, the systematic deviation of the real surface from the ideal surface. The remainder of the deviation of the real surface from ideal form is treated as a narrow zone

of uncertainty about this model surface, derived from the residuals of the sparse data about it. This model surface, as embellished by the uncertainty zone, is then treated in exactly the same way a dense set would be treated by a minimum zone method in establishing the ideal substitute geometry. A schematic illustration of the concept of this "Extended Zone Method" for the case of a nominally planar surface with form deviation modelled as a low order polynomial is given in Figure 1. In this presentation the planar case will be our focus in illustrating the method.

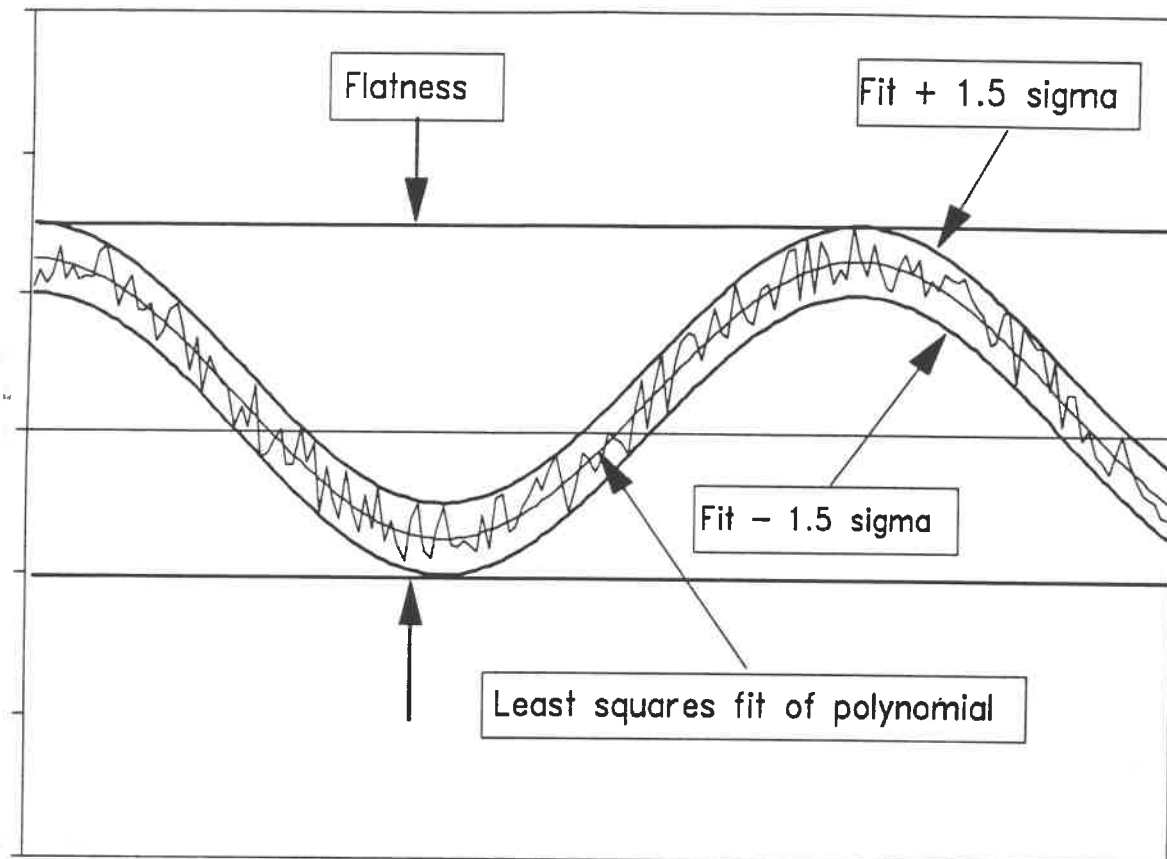


Figure 1. Schematic of Extended Zone Method

Generally the model surface is a linear combination of a set of basis surface shapes. Key to our method is the determination of what basis set is appropriate for a given production method. This involves dense sampling of sufficient part surfaces to establish the range of variability of the process. Our work has found that in some instances a basis set of relatively simple, smoothly varying analytical functions can adequately model the measured surface data. For nominally planar surfaces, an appropriate model can often be constructed as a product function of polynomials of relatively low order in x and y . We have used the Chebyshev polynomials because they offer several computational advantages, but this is not a critical issue. For nominally cylindrical surfaces, the axial dependence of the radius (r) can also be treated adequately by a low order polynomial in z , while the angular (θ) dependence is best treated as a Fourier function. None of these however is adequate to represent systematic form deviations which include discontinuities such as the steps sometimes encountered in planes produced by milling operations. In these instances a very effective alternative to the exclusive use of analytical surfaces is the inclusion of "eigen-surfaces" derived from factor analytical treatment

of the dense data sets. Briefly one begins with a suite of M nominally identical part surfaces, each densely sampled at N (fixed) locations, resulting in a data matrix $\mathbf{D}$ of dimensions $M \times N$. The square symmetric product moment matrix is formed as $\mathbf{D}\mathbf{D}^t$ and diagonalized. This process yields a set of M orthogonal ($1 \times N$) "eigen-surfaces" and, for each, a corresponding eigenvalue proportional to that surface's importance in accounting for the variance in $\mathbf{D}$. The eigen-surfaces can be used in conjunction with basis surfaces of perfect geometric form which accommodate fundamental orientation and location characteristics of the model.

Once the appropriate basis set has been established for modelling the surfaces, *sparse* surface sampling locations can be selected which yield a minimum variance estimate of the model surface error over its entire domain. This is an exceedingly important aspect of our efforts and hence will be further developed here for the case of nominally planar surfaces. We begin with our surface model equation

$$\mathbf{z} = \mathbf{B}\mathbf{a} \quad (1),$$

where $\mathbf{z}$ is an n dimensional column vector which is the surface represented by a *sparse* set of z values. Each row in the vector corresponds to a single point on the surface with a location of (x, y) . There are n points on this surface. The n by m matrix $\mathbf{B}$ is a set of basis shapes for the surface. Each column of this matrix gives the shape of a single basis surface. Each row corresponds to the same x, y location as the corresponding row in the $\mathbf{z}$ vector. The m dimensional column vector $\mathbf{a}$ contains the coefficients of the fit. The form of $\mathbf{B}$ is

$$\mathbf{B} = \begin{bmatrix} 1 & x_1 & y_1 & f_{41} & \dots & f_{m1} \\ 1 & x_2 & y_2 & f_{42} & \dots & f_{m2} \\ \dots & \dots & \dots & \dots & \dots & \dots \\ 1 & x_n & y_n & f_{4n} & \dots & f_{mn} \end{bmatrix} \quad (2).$$

Here the first column of $\mathbf{B}$ is a surface of constant height. The second column of $\mathbf{B}$ is a surface which increases in height with x . The third column of $\mathbf{B}$ is a surface which increases with y . These three "shapes" allow the representation of an arbitrary perfect plane. The additional columns give other shapes, which may either be expressed mathematically by an equation or may be derived from an experimental result. Typically we might use, in three columns, the functions x^2 , xy , and y^2 , to represent an arbitrary quadratic (second order) surface. Alternatively we might chose in successive columns one or more eigen-surfaces from a factor analysis.

If we represent the *directly measured* z values by a column vector $\mathbf{z}_d$, the least squares solution is $\mathbf{a} = (\mathbf{B}^t\mathbf{B})^{-1}\mathbf{B}^t\mathbf{z}_d$. Under the assumptions that the errors in z_i are normally distributed, independent, and have the same standard deviation, σ , the variance of the parameter a_k is given by σ^2 times the k -th diagonal entry of the matrix $(\mathbf{B}^t\mathbf{B})^{-1}$. A key characteristic of the least squares fitting is that this variance does not depend on the values of z_i . The only influence from the z variable is through σ , which we assume is the same at all points. Hence the variance in the parameters can be minimized by adjusting the points (x_i, y_i) where the measurements are taken.

It is also possible to estimate the variance for the predicted value of *any* z_k , given the x_k and y_k values. Consider the row vector $F'(x_k, y_k)$ given by

$$F'(x_k, y_k) = [1, x_k, y_k, f_4(x_k, y_k), \dots] \quad (3).$$

This is the vector of our basis sets evaluated at the location (x_k, y_k) . Then the predicted value is given by the formula $z_k = F'(x_k, y_k)a$. The estimate of the variance of the prediction for z_k is

$$F(x_k, y_k)(B'B)^{-1}F'(x_k, y_k). \quad (4).$$

The parameter we choose to minimize to find the optimal point *sparse set* is the average of this variance over all the points of the *dense set*. If we now restrict the region where the points may lie, we are led to a constrained optimization problem: find points $\{(x_i, y_i)\}$ of the *sparse set* that minimize the average of the variance of the prediction over the *dense set*. The average variance is a function of the points of the *sparse set* through the matrix B . We have written a numerical procedure for this problem. Given an order of the fit (the degree of the Chebyshev polynomials to use) a number of points to be located, and a *dense set* of potential point locations, the procedure finds where to locate the points to minimize the average of the variance over the region (as defined by the domain of the *dense set*). Thus the set selected will always be a subset of the proposed *dense set*. This approach then allows surfaces of arbitrary complexity in boundaries and internal voids to be handled with ease.

These methods have been used to study a wide variety of part surfaces in lot-size quantities. The parts have been densely sampled and these data have been processed by a variety of standard algorithms to yield feature (primary datum) location, orientation and form values. The success of the new methods for sampling and for analysis have been gaged by comparing these values with their counterparts obtained from optimized sparse sampling of the same parts and treatment of the sparse data by both previously established and new algorithms. For easy comparison, an *rms* "composite error" parameter is defined over the set of parts for each pattern/analysis method used. A composite error, expressed in linear units, is determined, for each part in the lot studied, as the largest of (1) the datum location error, (2) the direction cosine $i \times$ the part's x dimension, (3) the direction cosine $j \times$ the part's y dimension, (4) the flatness error.

We have found that, compared to *uniform* sampling patterns and standard algorithms for analysis, our sampling patterns and analysis methods can generally reduce *rms* composite errors by fifty percent and often by more than that, especially for quite small numbers of sample points. It is interesting to note also that our sample point optimization methods seem to suggest locations where the systematic form deviations are at or near their extremes. As a consequence of this, even the method of least squares and the minimum zone method yield better accuracy when our optimized point patterns are used rather than uniform sampling patterns.

**Work supported by the Quality Assurance Program of the Institute for Manufacturing and Automation Research, Consortium for Advanced Manufacturing International, Arlington, Texas.*

RECENT ADVANCES IN STYLUS BASED 3D SURFACE MEASUREMENT AND CHARACTERIZATION

Dr Paul J. Scott* & ** & Dr Euan Morrison*

* Rank Taylor Hobson Ltd
2 New Star Road
Leicester LE4 9JQ, UK

** School of Manufacturing and Mechanical Engineering
The University of Birmingham
Edgbaston
Birmingham B15 2TT, UK

INTRODUCTION

Currently, there is an increasing demand from industry to measure and characterise surfaces in 3D. This is becoming particularly important in applications where the surface texture of a component is critical in determining its functional properties (e.g. wear, lubrication, sealing or adhesive characteristics). Traditionally, the characterization of engineering surfaces has been carried out using single line profile measurements obtained from stylus based instruments.

Although these devices can be adapted to perform 3D surface measurement, the relatively slow traverse speeds of the stylus profilometers result in long measurement times, even when examining small surface areas. This has, in general, prevented 3D surface characterization from being used for inspection and control purposes in production environments.

Recent advances in the dynamic performance of stylus profilometers, made by one of the authors [1], has opened the possibility of high speed surface measurement using a stylus. As a result, a novel compact scanning profilometer - Talyscan - has been developed which is capable of rapidly measuring small surface areas. A description of the principle of operation of the instrument, including the dedicated 2D scanning traverse mechanism and fast acting stylus gauge, will be discussed.

Due to the compact design of the instrument and its ability to acquire large amounts of data in a relatively short period of time, this instrument has the potential for addressing a number of production environment measurement problems. However, in order to effectively introduce 3D surface characterisation into manufacturing environments, there is also an increasing requirement for additional surface characterization and analysis techniques over those which are currently used with single line (2D) profiling instruments. Many proposed 3D characterizations are just simple extensions of profile parameters and do not reflect the true 3D nature of a surface.

In order to illustrate this point, consider the surface shown in figure 1. This is a measurement of a 0.5mmx0.5mm square area on the surface of a plateau honed cylinder liner, performed using the Talyscan instrument in just 50 seconds. In terms of its function, the surface of the cylinder liner must be capable of withstanding wear, retaining lubricating fluid and providing some degree of sealing. It is clear that the complex nature of the surface cannot be fully characterised by performing a single 2D line profile measurement.

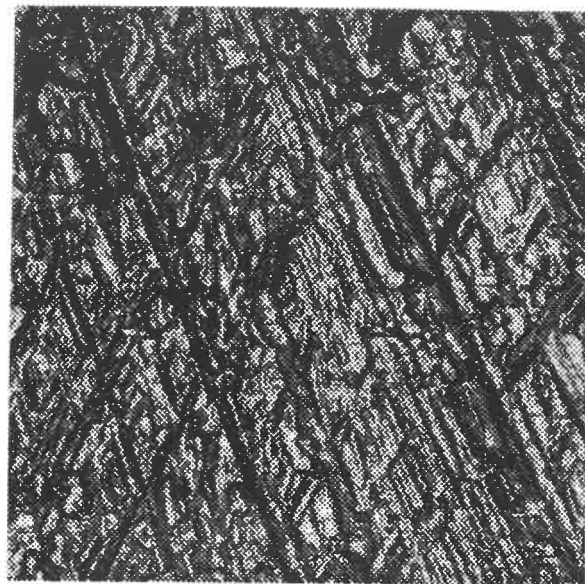


Figure 1 Plateau honed surface

Most current definitions of areal surface texture parameters are statistical in nature. Examples taken from the recommendation given in a recent BCR report (ref) on areal surface texture include :- RMS deviation of the surface S_q , Skewness S_{sk} , Kurtosis S_{ku} , Density of summits S_{ds} , Mean summit curvature S_{sc} etc.

Morse [2] has demonstrated that a knowledge of the topology of a surface can be of considerable help for many applications. The structure of the surface can be described by selecting a small set of "surface specific" points and lines that convey more information about the surface than just their co-ordinates. This approach is explored further in the next section.

TOPOLOGICAL CHARACTERIZATION

There is one discipline that has been measuring surfaces for thousands of years, namely cartography. It has built up an array of techniques to characterize surface features. The following is based on established techniques from cartography that characterize the topological features of a surface.

More than a hundred years ago Maxwell [3] proposed dividing a landscape into regions consisting of hills or alternatively consisting of dales. A hill is a set of points from which maximum uphill paths lead to one particular peak, and a dale is the set of points from which maximum downhill paths lead to one particular pit. By definition the boundaries between hills are course lines (watercourses), and the boundaries between dales are ridge lines (watersheds). Maxwell was able to demonstrate that ridge and course lines are maximum uphill and downhill paths emanating from saddle points and terminating at peaks and pits.

By combining hills and dale a useful representation of the surface can be obtained that incorporates peaks, pits, saddle points, course and ridge lines. The region where

a particular hill and a particular dale intersect is call a slope district. From all points within a slope district the maximum uphill and downhill paths lead to one particular peak and one particular pit respectively.

Pfaltz [4] introduced the concept of a weighted surface network, now call a Pfaltz graph, to represent the topological structure of a landscape. A Pfaltz graph is a weighted tripartite directed graph consisting of three sets of points, (namely the set of peaks, the set of saddle points and the set of pit points), together with a set of directed edges consisting of edges from a pit to a saddle corresponding to a course line, edges from a saddle to a peak corresponding to a ridge line and edges between saddle points corresponding to ridge-course lines. The edges are weighted according to the attributes of interest, usually this is the height difference between the two connected objects.

The importance of Pfaltz graphs is not just for their applicability to describe the topological structure of the surface but rather from the fact that they can be simplified in a way which preserves the significant topological structure. Wolf describes this process of simplification by combining edges with vertices (called w-contractions) and presents an algorithm based on this process. This simplification process can be used to eliminate insignificant features from a surface and is the areal equivalent of motif combination used on a profile.

The Pfaltz graph with appropriate simplification is the mathematical representation of the surface from which topological characterization is calculated and represents a powerful tool for the formal analysis of topographic surfaces. In general the theory from cartography provides a language and a structure to discuss and analyze surfaces.

CONNECTABILITY OF VALLEYS

One important example of a true areal characterization is the connectability of surface valleys. This is important for applications such as lubrication and paintability of a surface. This example will be used to illustrate how the use of the Pfaltz graph representation can overcome many of the problems in defining, selecting and classifying functionally significant features.

Kweon & Kanade [5] introduced the concept of a topographic change tree to describe the conectability of a surface. The change tree represents the relationships between contour lines from a surface. The vertical direction on the change tree represents height. At a given height all individual contour lines are represented by a point which is part of a line representing that contour line continuously varying with height. Saddle points are represented by the merging of two or more of these lines into one, peaks and pits are represented by the termination of a line.

From a change tree it is very easy to deduce the conectability of the surface. Unfortunately in practice a change tree can be dominated by very short contour lines, due to noise etc., which hinders interpretation. Fortunately change trees can be

calculated from Pfaltz graphs directly. By using an appropriate simplification procedure on a Pfaltz graph before calculating the change tree, the change tree is simplified and still contains the relevant connection information.

Finally given the change tree it is fairly straight forward to calculate other attributes associated with connected valleys such as:- the volume of the connected valleys, their direction of connection and distribution over a surface etc.

VISUALIZATION

Visualization techniques are among the most valuable for surface texture inspection because the image "looks" like it would be expected to, had the observer's eyes been sensitive to the information bearing media. Consequently, all knowledge built up over a life time of visual experience is brought to bear; it is indeed true that a picture is worth a thousand parameters. Regardless of the nature of the imaging species, stylus, visible or non visible light, electron, X-rays etc. an image conveys a maximum amount of information in a minimum time.

The vast amount of information conveyed in an image makes visualization an important diagnostic tool against the unexpected. It is envisaged that visualization will mainly be used in the research, feasibility and initial setting up stages of a production process. Once a process has been established and is stable, quantitative rather than qualitative information is required for process monitoring. This is partly due to both the increasing requirement to deskill metrology and to the use of quality control techniques such as SPC. Hence visualization techniques complement texture parameters in their areas of applicability and their usage.

Finally techniques for visualization of surfaces will be examined. Techniques from simple grey scale images to full ray tracing models and from random dot stereogram to animation flybys will be illustrated. No one technique is better than another, the strengths and drawbacks of the various visualization techniques presented will be discussed.

REFERENCES

- [1] Morrison, E.:(1995), *A prototype scanning stylus profilometer for rapid measurement of small surface areas*, Int. J. Mach. Tools Manu., **35**, pp 325-331.
- [2] Morse, S.P.:(1968), *A mathematical model for the analysis of contour lines data*, J. Assoc. Comput. Mach., **15**, pp 205-220.
- [3] Maxwell, J.C.:(1870), *On hills and dales*, The London, Edinburgh, and Dublin Phil. Mag. and J. Sci., Series 4, **40**, pp 421-425.
- [4] Pfaltz, J.L.:(1976), *Surface networks*, Geographical Analysis, **8**, no 2, pp 77-93.
- [5] Kweon , I.S. & Kanade, T.:(1994), *Extracting topographic terrain features from elevation maps*, CVGIP: image understanding, **59**, no 2, pp 171-182.

EFFICIENT CHARACTERIZATION OF SURFACE TOPOGRAPHY IN CYLINDER BORES

P. Pawlus*, D. G. Chetwynd

Department of Engineering,
University of Warwick, Coventry CV4 7AL UK

*permanent address: Rzeszow University of Technology,
ul. W Pola 2, 35-959 Rzeszow, Poland

There are good practical reasons for continuing to use stylus (contact) instruments for measuring surface roughness on engineering components. Since conventional profiling instruments cannot fully represent functionally important topographical features, mapping systems (or 3D roughness measurements) are becoming more common. There are no fully agreed standards for 3D characterization and so obvious extensions of profile parameters are being used. This begs the questions of whether such parameters are functionally significant and of whether typical profile measurement schemes are appropriate when mapping. Also, while a true surface map gives the highest quality information, a hierarchy of schemes is plausible in the trade-off of quality against measurement costs: mapping with close-spaced samples in each axis or with one axis sampled sparsely; parallel profiles at known spacing but without true vertical register; independent profiles in two orthogonal directions; independent profiles in only one direction.

Academic sources and instrument manufacturers tend to recommend relatively small data-sets for mappings, with a 256 by 256 point grid being at the upper end of acceptability. Computer power, memory and time costs cannot be neglected, but these restrictions appear to relate primarily to the slow rates of data collection from stylus instruments. Since surfaces often have large bandwidths of functionally significant topography, selecting the largest permissible sample interval is important in ensuring that an adequate area of the surface is measured by these restricted maps. A proposed method [1] for assessing this spacing for isotropic surfaces uses areal cumulative power spectra (that is the frequency axis integral of the power spectral density) to detect the lowest spatial frequency that includes a suitable arbitrary fraction of the total power (say 95%). Then, from the Nyquist criterion, the sampling rate is set at twice that frequency. It is not clear how best to treat highly anisotropic surfaces or ones that have highly skewed amplitude distributions. Hence we here examine the efficiency of various methods of measuring honed internal combustion engine cylinder liners since they are an economically important class, they have both forms of asymmetry and they are internal cylindrical surfaces for which precise setting up of instruments is generally more time-consuming than for external, nominally flat surfaces. Cross-measurements have been made on several instruments and under different conditions of six grey cast-iron cylinder liners honed in different ways. Together they represent a typical range of practical surfaces, with four having the classic plateau honed structure of a fine upper structure, providing a good bearing surface needing little 'running in', with an imposed cross-hatch pattern of deep grooves, which provide for oil distribution and retention.

Nominally flat surfaces are usually mapped by traversing one linear axis at approximately uniform speed, while sampling the probe amplitude regularly, and then stepping an orthogonal linear axis before recording the next track. Both axes are assumed to be sufficiently precise that

the vertical track-to-track misregister is much smaller than the shape errors of the surface. The orientation of the scan and step axes can be chosen for convenience. With cylindrical surfaces, the only sensible approach is to have a linear scan axis nominally parallel to the cylinder axis with the orthogonal scan a rotation centred nominally on the cylinder axis. It is aligning this rotary axis that increases set-up cost over that than for planar scanning. Kinematically, mapping by scanning the rotary axis and stepping the linear one or *vice versa* should be identical if the sampling interval is the same in both directions. Practical imperfections, including residual misalignment, mean that the equivalence is not exact. Thus an initial study compared parameter values obtained from each configuration. The rotational scan was achieved by modifying a Rotary Talysurf (Rank Taylor Hobson Ltd. based on Talysurf 4) to allow a monitored variation of the height of the pick-up. The rotational axis was oriented vertically. The linear scan version used an RTH Talysurf 5 with the cylinder liner held on a conventional dividing head (horizontal axis) to provide the rotary steps. The systems were only semi-automated for this single experimental study, with the stepped axis controlled manually. Scans were at nominally constant speeds typical of normal surface roughness measurement and data were sampled at constant time intervals into a personal computer, using a 12-bit analog to digital converter. The resolution in the vertical axis was approximately 6 nm. Sampling intervals were 10 μm in each axis with 128 parallel scans of 256 samples, giving an assessment area of 2.56 by 1.28 mm. A linear least squares best fit plane was fitted through the data set prior to the calculation of the parameters.

The parameters assessed were average and peak-to-valley amplitude, skewness, kurtosis and the ratio of peak amplitude to the peak-to-valley, all taken over the whole data set. Amplitude parameters tended systematically to be slightly higher when scanning in the rotary axis, while variations in the other parameters were more scattered, but differences were generally less than 10%. Such differences could be expected from different samplings of one specimen and exact correspondence between instruments is not to be expected: calibration uncertainty and actual stylus size and shape will cause some variation (both instruments had nominally 2 μm pyramid tips). There is thus no evidence that scanning in the rotary axis is beneficial. This is welcome, since this arrangement is the more complex to use in practice.

An even simpler instrument arrangement arises if a planar (x,y) scan can be used with a stylus pick-up having enough range to accommodate the curvature of the cylinder over a sufficient area. The penalty paid is that the measurement axis is not consistently orthogonal to the surface and a curved reference figure must be fitted to the data. A set of tests was run on an RTH Form Talysurf fitted with a y -stepping table and having these capabilities to compare it to the linear scan, radial step configuration. The internal software of the Form Talysurf prevented close matching to the previous test regime and a 164 square set of samples at 30 μm spacing was selected. Parameter differences were again rarely more than 10% with samples taken from similar areas of individual liners and no consistent trends were found. The variations are well within those expected from differences in stylus shape, sampling and residual calibration errors. The use of planar scan instruments appears to be quite acceptable, at least over sample areas of a few millimetres lateral extent, in the presence of curvatures typical of smaller cylinder liners.

The cumulative (or accumulated) power spectrum method of estimating sampling intervals for nearly isotropic surfaces [1] is itself time-consuming. Both its applicability to asymmetric surfaces and the potential for estimating sampling from spectra of *profiles* taken in the axial (easy to measure) direction are of interest. Simulated, ideal, data sets showed that the mixed topography of plateau honed surfaces causes difficulties: the cumulative spectrum does not

rapidly converge to a suitably high value because of the very different spectral content of the deep scratches and the plateau roughness. However, this can be exploited since the cumulative spectrum shows a distinct 'knee' with an almost linear upper region. A simple line fit to this part allows the divergence at the knee to be detected readily and this point can be used to estimate the sampling interval. Although only experience gives a criterion for detecting the knee (deviation from the line of 2% of total power proved good here), the procedure is not totally based on an arbitrary measurement value but has some functional significance built in.

A different, novel approach to estimating best sampling intervals for plateau honed surfaces, based on the concept of deep valley analysis [2], was also tested. Essentially it identifies valleys from a close-sampled measurement as 'deep' through criteria derived from the bearing curve and establishes their typical width. The 'best' sampling interval is then chosen so that, averaged over all alignments of the sample points relative to the scratch, the detected depth exceeds some precept value. For example, if the scratch is idealized to triangular form, requiring an average 75% of true depth implies a spacing of half the full scratch width.

Experimental testing of both approaches used measurements taken on the Talysurf 5 based system. Six cylinders were measured with 256 point axial scans in 128 parallel tracks, equisampled in both axes at each of 5, 10 and 20 μm . The starting point of the measurement was the same for each interval. Additionally, ten independent axial profiles were taken with 5 μm samples over 5.12 mm traverses, corresponding to the largest dimension of the area measurements. As a cross-check, independent profiles sampled at 5 μm over 2.56 mm traverses were taken circumferentially using the Rotary Talysurf.

The deep scratch widths along axial profiles were in the 30 to 50 μm range, while circumferential profiles indicated typically 2 to 3 times that size, consistent with grooves angled at around 60-75° to the axis. Sampling intervals were proposed as 0.3 of these widths. Evaluating deep scratch depths [2] in each direction for the different sampling intervals confirmed that the proposed sampling was plausible. Agreement was good between samplings at 5 and 10 μm , but in some cases large differences were found with 20 μm sampling. The divergence was strong when the proposed sampling was below 20 μm and moderate when it was close to that value. Determining the true valley width requires profiles in two (preferably orthogonal) directions, but the present evidence indicates that generally satisfactory results may be obtained by using the nominal honing angle, if known, to scale the width detected in axial profiles alone.

Sampling intervals proposed from the cumulative spectrum method were generally similar to those from the deep valley method, tending to be slightly larger, but never by more than 2 μm (in values ranging from 10 to 30 μm) in the axial direction and 3 μm circumferentially. It was lower by 1 μm in two cases, for both of which the scratch depth data was moderately variable in the critical region of sampling. Computations of the power spectra at all the measured sampling intervals confirmed that little power was missed provided that the sampling did not exceed the proposed values. Further, the absence of a distinct knee acted as a diagnostic of inadequate sampling. Figure 1 shows cumulative spectra of the same specimen obtained at 5 μm , 10 μm and 20 μm sampling. From the close-spaced spectrum, sampling ought to be at about 10 μm . The axes for (a) and (b) indicate the full spectral range of each measurement and show that the proposed sampling spreads the data well across that range. Computations of amplitude and shape parameters also highlighted significant divergence at samplings above those proposed (except for averaged amplitudes which are almost insensitive to aliasing effects).

We conclude from this study that similar quality of data is obtained from different configurations of mapping instruments and that the simplest may be used for cylinder liner measurement. The original method of calculating sampling from cumulative power spectra is inadequate for plateau honed surfaces, but a simple modification introduced here rectifies that difficulty. A somewhat simpler approach based on the analysis of deep valleys may also be used. Both methods appear to be practicable but the modified cumulative power spectral approach may be slightly more precise. It is certainly not inferior and should therefore be preferred if only because it can be applied to other types of surfaces and so offers greater utility. Finally, it appears that adequate information about most features (but not, for example, honing angle) can be obtained from profiles alone by using reasonable assumptions based on usually well-controlled parts of the manufacturing process.

References

- [1] TY Lin, L Blunt and KJ Stout, Determination of proper frequency bandwidth for 3D topography measurement using spectral analysis. Part I: isotropic surfaces. *Wear* **166** (1993) 221-232.
- [2] J Michalski and P Pawlus, Description of honed cylinders surface topography, *Int. J. Mach. Tools Manufact.* **34** (1994) 199-210

Acknowledgment

We are grateful to Dr J Coy, GKN Wolverhampton, for offering advice and facilities and to the British Council for their support of Dr Pawlus' work at Warwick University.

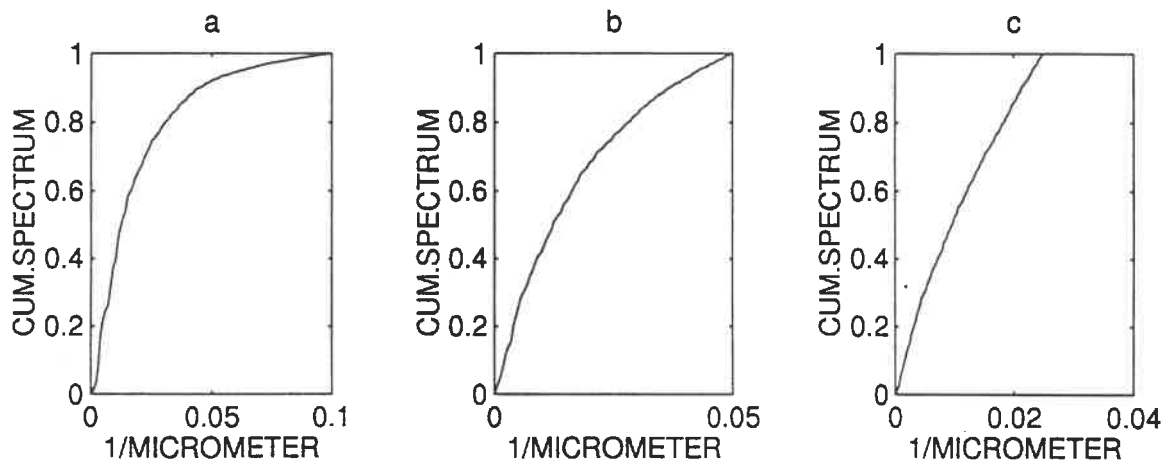


Figure 1: Typical cumulative power spectra of a plateau honed cylinder liner sampled at 5, 10 and 20 μm , where the sampling interval proposed by the new criterion is 10 μm .

APPLICATION OF CONFOCAL SCANNING OPTICAL MICROSCOPE TO THREE DIMENSIONAL SURFACE AND FORM ERROR CHARACTERIZATION

Ganesha Udupa*
Research Scholar

M.Singaperumal *
Assoc. Professor

R.S.Sirohi @
Professor

M.P.Kothiyal @
Professor

*** Precision Engineering & Instrumentation Laboratory,
Department of Mechanical Engineering.**

**@ Applied Optics Laboratory, Department of Physics.
Indian Institute of Technology
Madras-600 036, India.**

Introduction

Surface geometry and in particular texture and form have played an important role in determining the performance of engineering parts. Form errors of three-dimensional surfaces like cylindrical and spherical surfaces are important from the point of the function they perform in an assembly. Hence evaluation of form errors at different zones on the component as well as form error characterization of the component is essential.

The present work reports on the development of an optical method based on confocal scanning optical microscope[CSOM] for accurate measurement and assessment of cylindrical form errors. The instrument measures both surface and form errors. The form error namely roundness and cylindricity are measured by rotating the object in a V-block. A simple reversal method [1] was used for separating the workpiece error from the runout error of the workpiece. It is found that by measuring the lobed workpieces outside the V-block support gives less runout error of the workpiece. The roundness and cylindricity profiles can then be drawn with correction for runout error on the measured data. Least square technique is used as a reference datum for roundness, cylindricity and 3-D surface roughness evaluation. A software perform the data processing, analysis and computation of 2-D and 3-D parameters to characterize the measured component.

The 3-D profile measuring System

A non contact profile measuring system is developed to assess the 3-D surface topography and form error of cylindrical components by CSOM based on the focus detection technique. Fig 1. shows the schematic diagram of the measuring system. A focussed laser spot ($\approx 0.6 \mu\text{m}$ dia) through a microscope objective falls on the surface of the cylindrical part which is rotated by a computer controlled stepper motor placed on a motion controlled x-y table. The workpiece, placed on the V-block is rotated freely by the motor through a set of gears which provides 400 data points to be measured around it. The focused laser beam scans radially with the help of a piezo electric translator(PZT). The maximum intensity of the reflected light from the cylindrical surface at a point is detected by a photodetector. The movement of the PZT corresponding to the maximum intensity of the reflected light is recorded as the radial displacement of the cylindrical surface in case of roundness measurement or the vertical dimensional information of the surface topography. The radial displacements or the data are obtained at different horizontal sections and the required form error is then evaluated using the least squares technique. A software controls the movement of PZT for the

radial positioning of the laser light. PZT's with vertical resolution of $0.040\mu\text{m}$ and $0.080\mu\text{m}$ were used for the experiment. The instrument is calibrated using standard ground pieces.

Characterization of circularity and cylindricity

In general, profiles have to be characterized in two directions. The roundness profile has to be characterized in both the radial and circumferential directions and the cylindricity profile has to be characterized along radial and longitudinal directions. The radial characteristics of the roundness profile are assessed in terms of the roundness error O , the maximum peak height P , the maximum valley depth V , the mean line average height M and RMS value. The circumferential and longitudinal characteristics of the profile are given by the average undulation number and the average wavelength index respectively. These parameters provide information about the nature of the surface and give a quantitative basis for specifying the openness and closeness of the profile.

The cylindricity is assessed by the cylindricity error (C_{p-v}) which is the sum of the maximum and minimum deviations from the least squares cylinder which is used as the reference cylinder for measuring cylindricity. The main limitation in the reference cylinders is that the cylindricity peak to valley parameter gives no indication of the 3-D form of the component. To overcome this, a new set of parameters is required to distinguish form from peak to valley error. Here some of the parameters are suggested which will be useful in determining the shape of the components. Apart from the slope and axial straightness error, volume calculation of the cylindrical components helps in determining the shape of the component having same peak to valley error and length of the cylinder. Some of the 2-D or 3-D surface roughness parameters can be extended to characterize roundness and cylindricity measurements. Some of these parameters are the average height C_s , RMS value C_q , RMS slope of the cylinder $C_{\Delta q}$ and some functional parameters for characterizing bearing and fluid retention properties. This requires cylinder bearing area ratio curve to calculate cylinder bearing index C_{bi} , core fluid retention index C_{ci} , valley fluid retention index C_{vi} (corresponding to 3-D surface topography parameters). For a specific functional application such as in an assembly having clearance or interference fit like those in hydraulic pistons and bearings, frictional parameter will be useful in describing the characteristics of the surface such as wear, friction, lubrication, sealing, loaded area etc. This parameter can be obtained from the measurement data and the profile graph of the cylinder by knowing the coefficient of friction and the mean diameter of the piston and cylinder.

Characterization of 3-D surface topography

The 3-D topographic data was collected using the developed optical instrument. The areas to be mapped are identified by markings over the surface. An area of $2 \times 2 \text{ mm}^2$ was mapped which provides a superset of data from which smaller areas and individual can be sampled. Sample interval of $10 \mu\text{m}$ is selected by the software. For area assessment the size of the subset (window) chosen for the characterization was $0.8\text{mm} \times 0.8\text{mm}$. This is the 3-D equivalent of the conventional cut-off length chosen when analysing surfaces using 2-D profilometry. 3-D parameters can be calculated from the measured height data obtained from the subset [2]. The window can be moved over the surface to provide an ensemble of area parameters. An overlapping window with a separation of $50 \mu\text{m}$ was selected for our experiment. For calculation of 3-D profile parameters a least squares mean plane is used to provide a suitable characterization datum.

Results

Experiments are conducted on steel cylindrical components with and without lobbing errors. The 3-D topographic features of the various machined and optical surfaces are also measured. In

case of roundness measurement, the separation of workpiece error from the runout error of the workpiece requires atleast two roundness profiles to be measured by changing the orientation of the workpiece on the V-block. Fig. 2 shows the V-block method of measuring 3-lobed workpiece. Fig. 3 and Fig. 4 shows the polar diagram representation of the lobed workpiece obtained by CSOM and Perthan form tester. Calculated values of the tested lobed and centerless ground cylindrical workpiece are listed in table 1. The out of roundness measurement by CSOM closely agrees with that of stylus method. The roundness measurements were taken at different sections of the cylinder at equal intervals for a length of 30 mm for calculating cylindricity error.

The 3-D surface topography of different machined and optical surfaces are measured using CSOM. Fig. 5 shows the 3-D topographic features of a ground surface. The calculated 3-D parameters of some typical surfaces are listed in table 2. The values of parameters obtained by CSOM agrees with that of stylus instrument. The relationship between various parameters are established using linear regression analysis. Slope and harmonic analysis were carried out which will be useful in controlling the manufacturing process.

TABLE 1

Process	Device	P *	V *	O *	M *	h_q *	P_{av} *	V_{av} *	C_{p-v} *
3-lobed (milled)	CSOM	7.43	-5.55	12.99	5.965	6.922	3.235	-3.232	13.58
	Stylus	7.42	-5.50	12.83					13.52
centerless ground(CG)	CSOM	1.18	-1.04	2.22	0.305	0.086	0.295	-0.224	5.25
	Stylus	1.17	-1.04	2.21					5.23
		N	λ_{av} *	$h_{\Delta q}$ **	Max slope	Min slope	Avg slope		
3-lobed	CSOM	3	0.305	0.158	0.223	-0.22	0.142		
CG	CSOM	12	0.115	0.882	1.247	-1.24	0.778		

TABLE 2

Process	Device	Amplitude Parameters						Functional Parameters (μm)		
		S_a *	S_q *	S_t *	S_z *	S_{sk}	S_{ku}	S_{bi}	S_{ci}	S_{vi}
Ground	CSOM	0.413	0.610	2.25	9.94	-4.850	35.12	1.032	0.791	0.170
	Stylus	0.402	0.592	2.12	9.82	-4.825	34.87	1.021	0.785	0.168
Polished	CSOM	0.040	0.049	0.24	0.27	-0.008	3.582	0.755	1.245	0.145
	Stylus	0.038	0.048	0.23	0.26	-0.006	3.573	0.756	1.225	0.142

* μm ** $\mu m/\mu m$.

Conclusion

An optical method based on focus detection technique is employed for the evaluation of 3-D surface and form errors. A new method of measuring the form errors over a V-block is suggested. The values of the parameters obtained by CSOM are compared with those of stylus instrument and found to be in close agreement.

References

1. R.R. Donaldson , A simple method for separating spindle error from test ball roundness error. Annals of the CIRP, V21/1,1972, pp 125-126.
2. K.J. Stout (Ed), Three Dimensional Surface Topography; Measurement, Interpretation and Applications , Penton Press, London, 1994.

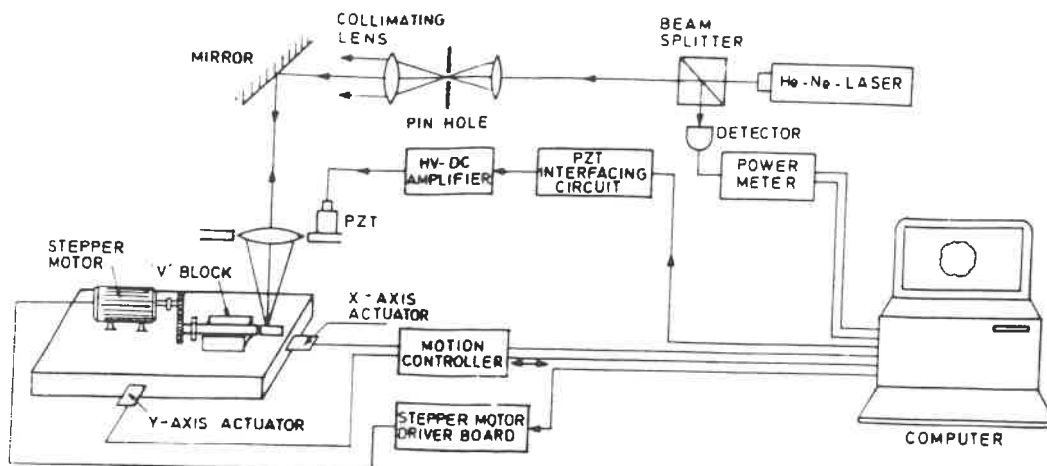


FIG. 1 A SCHEMATIC DIAGRAM OF CSOM FOR ROUNDNESS MEASUREMENT

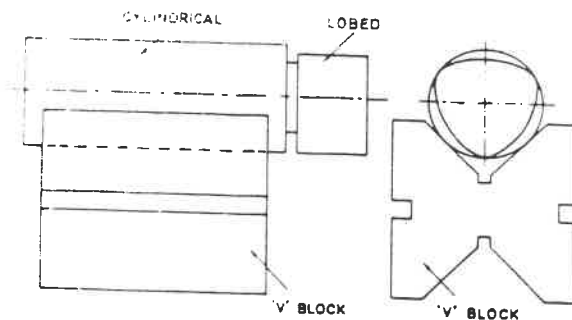


FIG. 2 V-BLOCK MEASUREMENT OF 3 LOBED WORK PIECE

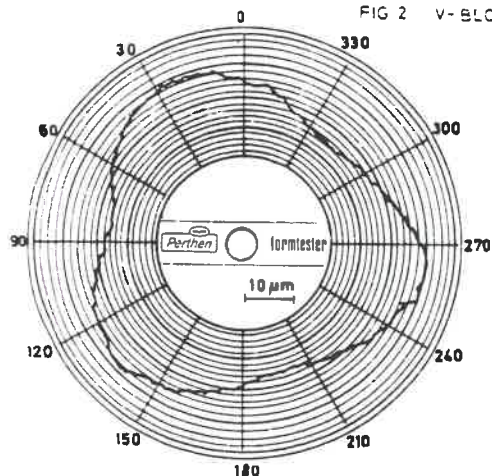


FIG. 4 PROFILE OBTAINED BY FORM TESTER

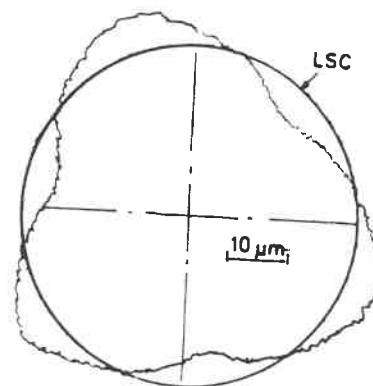


FIG. 3 PROFILE OBTAINED BY CSOM

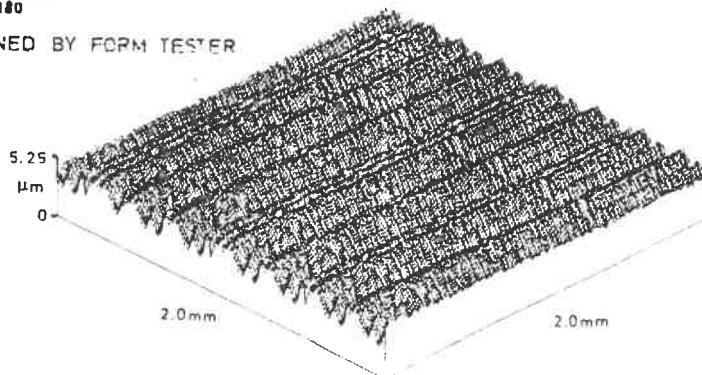


FIG. 5 THREE-DIMENSIONAL TOPOGRAPHIC FEATURES OF GROUND SURFACE

MODELING AND PREDICTION OF THERMALLY INDUCED ERROR MAPS IN MACHINE TOOLS USING A LASER BALL BAR AND A NEURAL NETWORK

Narayan Srinivasa
The Beckman Institute
University of Illinois at Urbana-Champaign
405 North Mathews Avenue
Urbana, IL-61801

John C. Ziegert
Department of Mechanical Engineering
University of Florida
Gainesville, FL-32611

Thermally induced errors are major contributors to the overall accuracy of machine tools. While this area is well researched, a major drawback of currently proposed methods is the amount of time consumed in building this model. Due to this drawback, there has been a reluctance in the industrial community to adopt these techniques. In this paper, a *direct* method of machine tool calibration is adopted to model and predict thermally induced errors in machine tools. This method uses a laser ball bar (LBB) as the calibration instrument and is implemented on a two axis computerized numerical turning center (CNC) turning center. Rather than individually measuring the parametric errors to build the error model of the machine, the total positioning errors at the cutting tool and spindle thermal drifts are rapidly measured using the LBB. The LBB is a linear displacement measuring device that consists of a displacement measuring laser interferometer aligned between two precision spheres by a telescoping tube as shown in Figure 1. The spindle thermal drifts and geometric errors of the machine axes are both measured within the same experimental setup using the LBB. These measurements adopt the sequential method of trilateration as shown in Figure 2. Unlike conventional approaches, the spindle thermal drifts are derived from the true spindle position and orientation measured by the LBB.

The thermally induced error map of the two axis turning center was measured using the LBB over a regular thermal duty cycle (TDC1) that simulated machining of large workpieces. TDC1 was divided into measurement and warming periods. During each measurement period, the LBB was used to rapidly measure the error map of the machine at various thermal states. A Fuzzy ARTMAP neural network was used to build a machine model in an incremental fashion by correlating the measured errors with temperature gradients of 14 points on the machine (refer to Figure 3(a) and 3(b)) during TDC1. The inputs to the network are the nominal position of the cutting tool and the temperature gradients. The outputs are the total positional errors of the machine. The inputs and outputs to the network are shown in Figure 4. The network parameters of the

model were chosen such that it behaves like an adaptive bin classifier where each bin represents a class that is an integral multiple of the machine resolution. The warming period of TDC1 consists of rapid traversing of both axes and simultaneously running of the spindle at a constant speed. The prediction ability of the network was then tested using a new thermal duty cycle (TDC2) that simulated machining of short workpieces with small setup times. The overall prediction ability of the network for TDC2 (not experienced by the network before) was found to be within $3\mu\text{m}$. The results for a couple of test trials are shown in Figure 5(a) and 5(b).

The network was then interfaced with a PC based error compensation system (ECS). The ECS breaks into the feedback loop between the position encoders of the machine axes and the machine tool controller and modifies the feedback to the controller by injecting or suppressing encoder pulses (equivalent to the predicted errors). The performance of the overall system was evaluated through cutting tests under various thermal states of the machine. Improvement ratios (uncompensated over compensated) of upto 13 in diameter, 9 in length and 15 in taper were obtained. A typical comparison between the desired, uncompensated, Fuzzy ARTMAP predicted and compensated part dimensions during the cutting tests is plotted in Figure 6. These results are encouraging considering that the prediction accuracy of the network was $3\mu\text{m}$ and the compensation resolution of the ECS was $4\mu\text{m}$.

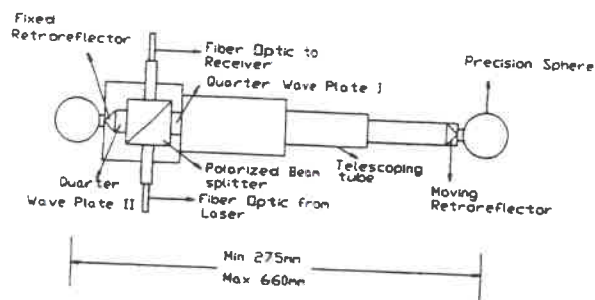


Figure 1. Schematic of the Laser Ball Bar.

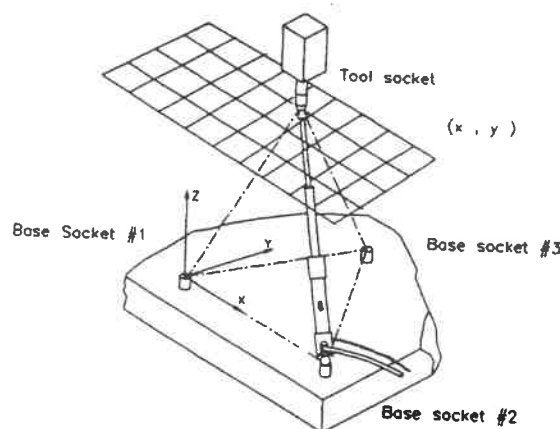


Figure 2. Sequential trilateration using the LBB.

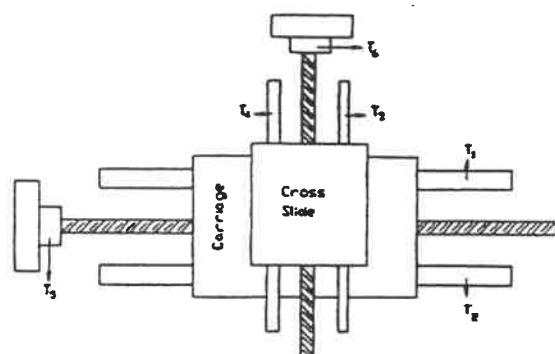


Figure 3a. Thermocouple locations on carriage and cross-slide elements.

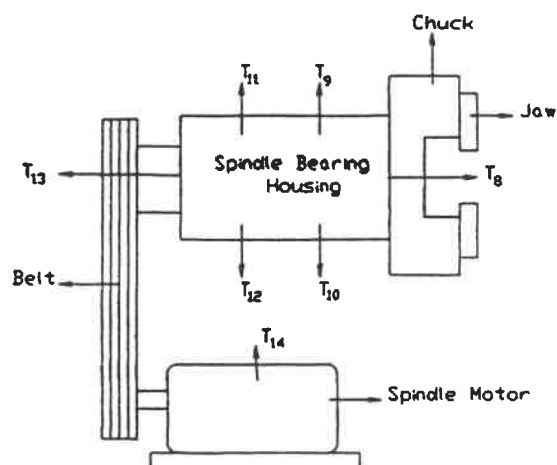
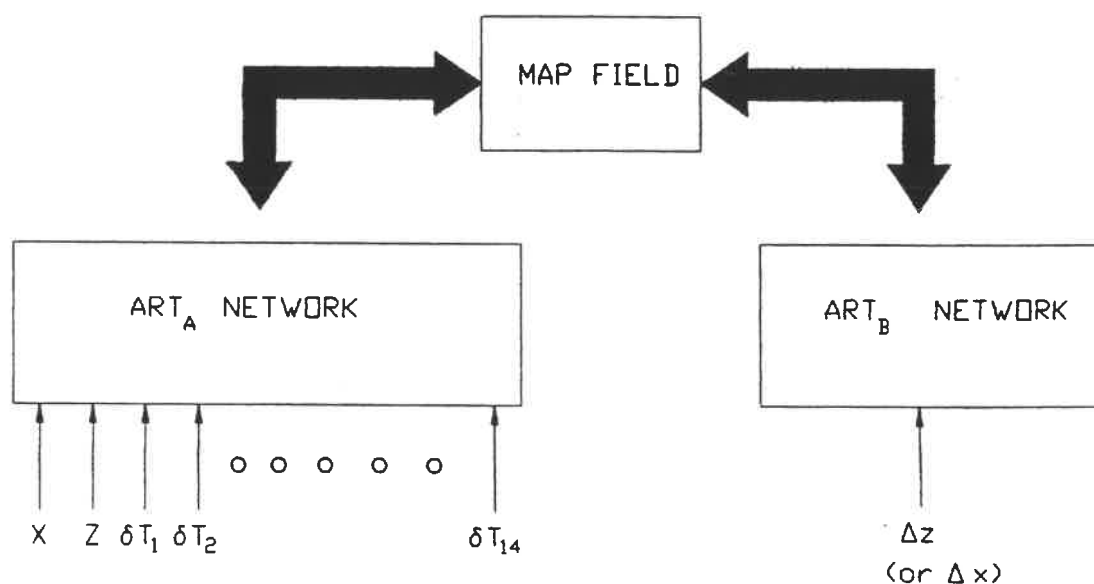


Figure 3b. Thermocouple locations on spindle elements.



x - cross slide position in machine coordinates

z - carriage position in machine coordinates

δT_i - temperature of i th heat source from its cold state

Δz - LBB measured total z positional error at cutting tool

Δx - LBB measured total x positional error at cutting tool

Figure 4. Inputs and outputs to the Fuzzy ARTMAP neural network.

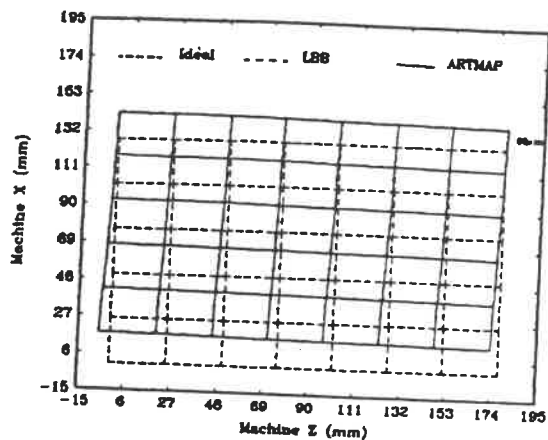


Figure 5a. Comparison between LBB and ARTMAP for trial#3 of TDC2.

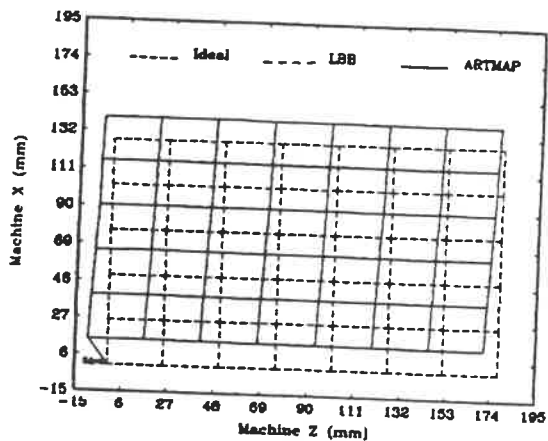


Figure 5b. Comparison between LBB and ARTMAP for trial #11 of TDC2.

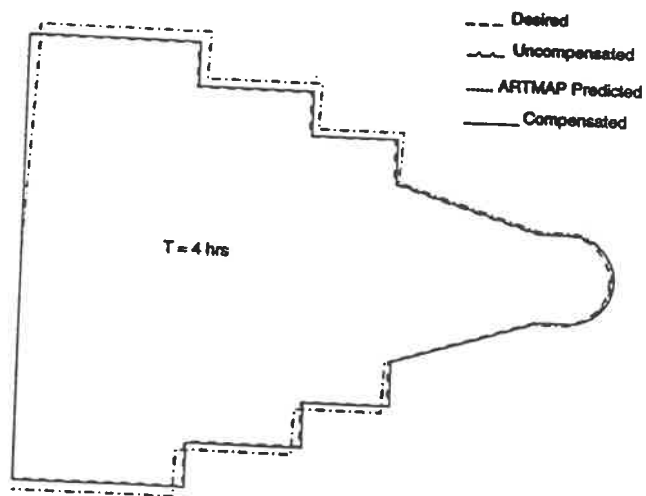


Figure 6. Comparison between desired, uncompensated, network predicted and compensated part dimensions.

RAPID MAPPING OF VOLUMETRIC ERRORS

Debbie Krulewich, Layton Hale, Don Yordy
Lawrence Livermore National Laboratory
Livermore, CA 94550

1. Abstract

This paper describes a relatively inexpensive, fast, and easy to execute approach to mapping the volumetric errors of a machine tool, coordinate measuring machine, or robot. An error map is used to characterize a machine or to improve its accuracy by compensating for the systematic errors. The method consists of three steps: (1) modeling the relationship between the volumetric error and the current state of the machine; (2) acquiring error data based on length measurements throughout the work volume; and (3) optimizing the model to the particular machine.

2. Background

There are three documented methods for modeling volumetric errors: (1) volumetric error modeling, (2) parametric error modeling, and (3) functional error modeling. Volumetric error modeling involves measuring the volumetric errors throughout the working space of the machine. The error at a given location is determined by interpolation between the measured points. For good accuracy, many points must be measured. Furthermore, to date an adequate measuring device does not exist. Parametric error modeling, the most commonly used method, measures the six parametric errors of each moving element and then combines them based on the geometry of the machine to form a functional point to workpiece error. Parametric error modeling is often not practical due to the complexity, time requirements, and expense. Functional error modeling represents the volumetric errors as a function of the current state of the machine

(commanded position, temperature, etc.), and the function is fit to volumetric error measurements. Because the model provides an intelligent way to interpolate between measured points, fewer measurements are required. Current technologies for acquiring volumetric error measurements include probing a known artifact, but it is difficult to acquire an adequate density of measurement points. Alternatively, we are using a new measurement technique based on length measurements for acquiring volumetric error measurements.

3. Model Development

The functional model is built from the parametric error model. The parametric errors are appropriately combined to predict the error between the functional point and workpiece. Mathematical expressions for each parametric error component are then substituted into the parametric error model. As a first attempt, we have used simple polynomials that are only dependent on commanded position. If no non-rigid coupling between axes is expected, the mathematical expression for each parametric error will only be dependent on the moving axis position; however, if non-rigid coupling is expected, the mathematical expression may contain other axes. It should be noted that the model development requires full knowledge of the machine behavior for a given machine design. However, the model is characteristic of all machines with that design so that the general model only needs to be developed once.

In the future, we will investigate other functional forms such as orthogonal polynomials and splines. We also plan to

This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under contract No. W-7405-Eng-48.

incorporate quasi-static thermal and load induced errors into the model. Along with the current location of the machine, the model would then include current temperatures of critical locations on the machine and the part weight.

4. Measurement Technique

A series of length measurements, taken from several fixed base locations to various functional points in the work volume, are compared with the respective lengths calculated between the commanded positions and the nominal base locations. The difference between measured and calculated lengths to a point is approximately equal to the component of error projected onto that measurement direction. This difference is then set equal to the projected error, expressed as a volumetric function. The parameters of the function are solved to optimize its global fit to all the measurements.

The notation in Table 1 will be used throughout. The error vector is defined as

$$\mathbf{e} = \mathbf{L}_m - \mathbf{L}_c \quad (1)$$

and the projection of the error vector onto the nominal commanded length vector is

$$\frac{\mathbf{L}_c}{L_c} \cdot \mathbf{e} = \frac{\mathbf{L}_c}{L_c} \cdot (\mathbf{L}_m - \mathbf{L}_c) = \frac{\mathbf{L}_c}{L_c} \cdot \mathbf{L}_m - L_c \quad (2)$$

Because the error is small with respect to the measured length, the projection of the measured length vector onto the commanded

length vector is approximately the magnitude of the measured length,

$$\frac{\mathbf{L}_c}{L_c} \cdot \mathbf{L}_m \approx L_m \quad (3)$$

Equation 2 then reduces to

$$dL \equiv \frac{\mathbf{L}_c}{L_c} \cdot \mathbf{e} \approx L_m - L_c \quad (4)$$

Table 1: Notations

(x_c, y_c, z_c)	commanded machine position
(x_p, y_p, z_p)	nominal base location
$\mathbf{e}$	error vector, (dx, dy, dz)
$\mathbf{L}_c$	nominal commanded length vector, $(x_c, y_c, z_c) - (x_p, y_p, z_p)$
L_c	magnitude of the nominal commanded length
$\mathbf{L}_m$	measured length direction (unknown)
L_m	measured length
dL	projection of $\mathbf{e}$ onto $\mathbf{L}_c$

Figure 1 illustrates the error projections in a two-dimensional plane. In this two-dimensional example, two lengths, L_{m1} and L_{m2} are measured from two nominal base locations, (x_{p1}, y_{p1}) and (x_{p2}, y_{p2}) , to the functional point located at $(x_c, y_c) + (dx, dy)$. The error vector $\mathbf{e}$, projected onto the commanded vectors, $\mathbf{L}_{c1}$ and $\mathbf{L}_{c2}$, yields lengths, dL_1 and dL_2 , that are approximately equal to the respective differences between the commanded and measured lengths.

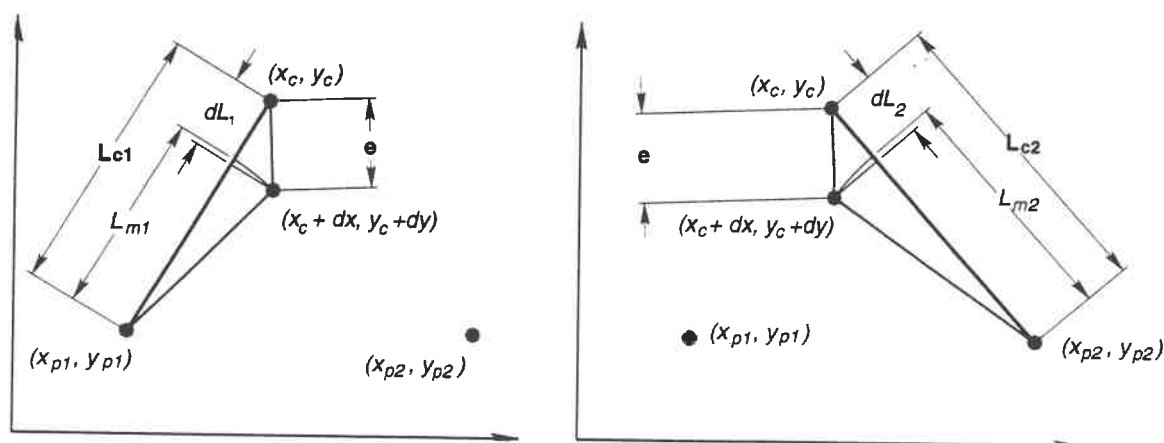


Figure 1: Simple 2-D Representation of Error Projections

The same reasoning applies to three dimensions, so that Equation 4 is completely general. Expanding Equation 4 for the three-dimensional case results in

$$\frac{(x_c - x_p)}{L_c} dx + \frac{(y_c - y_p)}{L_c} dy + \frac{(z_c - z_p)}{L_c} dz \approx L_m - L_c. \quad (5)$$

To solve for the three unknowns, dx , dy , and dz , in Equation 5, lengths from at least three base locations must be measured. Applying Equation 5 to only three measurements, one from each base location to a single point, would result in three linear equations that can be solved directly for the three-dimensional volumetric error. This is the linearized form of trilateration. This method of projecting errors onto the nominal commanded length vector, which we refer to as the error projection method, is naturally extendible to any number of measurements for the purpose of fitting the functional model. This is accomplished by substituting models for dx , dy , and dz into Equation 5. Then lengths from three or more base locations to the functional point are measured for commanded positions throughout the working volume. Applying Equation 5 to each commanded position results in an overdetermined set of equations that can be solved for the unknown model parameters using linear regression.

The non-systematic errors tend to be smoothed during the fitting procedure. If (1) the error model is correct and (2) the non-systematic errors appear to be random, the model fitting will have an averaging effect on the errors. To ensure that time dependent errors appear to be random, the measurement points are taken in a random order to disassociate time and position. For example, the measuring device, if accidentally warmed by the operator, will slowly cool causing measurement errors that are associated with time. If the measurements were taken in a particular order, for instance always increasing in the x direction, then the time

related errors will couple to the x -axis motion. Unavoidable time dependent errors can be reduced by randomizing the order of measurements so their random appearance will average in the fitting process.

5. Model Optimization

To ensure that the error model is correct, the general model for a given machine design must be optimized for the specific machine. The general model that describes all for a given design will have more terms than required for most machines. The extra terms hurt the predictive ability of the model because the extra terms begin to fit the noise in the data. Therefore, the best subset model must be selected for each particular machine. Unless the model is orthogonal, every possible subset must be considered. To find the best model, I have had most success using Mallows C_p statistic, which is an unbiased estimator of the total sum of squares of discrepancies (variance error plus bias error) of the candidate model from the true but unknown model, and is calculated by

$$C_p = \frac{RSS_p}{s^2} - (n - 2p) \quad (6)$$

where

RSS_p = residual sum squared error from a candidate model containing p parameters,

p = number of the parameters in the candidate model,

s^2 = residual mean squared error from the general model.

Since the expected value of C_p is p , adequate models will show up as points close to $C_p = p$. Using actual data, I have selected the best subset model and found that the optimized model fit an independently acquired data set better than the full model.

6. Results

The volumetric error map represents the systematic portion of the error, where the apparently non-systematic errors are those that are not taken into account in the model. The

degree of systematic error that can accurately be predicted is controlled by the number of measurements and the non-systematic errors of the length measuring device and the machine. During initial testing, we acquired 1500 length measurements throughout the volume of a three-axis machine in three hours using a Laser Ball Bar manufactured by Tetra Precision. Model predictions were then compared to independently acquired body and face diagonal errors measured with a laser interferometer. The model predictions were within 2 μm of the diagonal measurements, which is the combined non-systematic error of the measuring device and the machine. Given an infinite amount of data, the prediction should exactly predict the apparently systematic portion of the machine error assuming that the model is appropriate and

that the measurements are taken throughout the volume.

7. Conclusion

In conclusion, the functional error model can accurately represent the volumetric errors throughout the working space of a machine. Furthermore, the error projection method provides a means to recover volumetric errors from rapidly acquired length measurements. The reduction in time required to map the volume is the benefit of this method rather than improved accuracy over current technologies. However, the accuracy of the error map is comparable to that achieved using conventional methods. We have not explored the use of this method on ultra-precision machine tools and coordinate measuring machines. A length measuring device with adequate accuracy would be required.

1. Belforte, G.; Bona, B.; Canuto, F.; Donati, F.; Ferraris, F.; Gorini, I.; Morei, S.; Peisino, M.; Sartori, S. Coordinate Measuring Machines and Machine Tools Selfcalibration and Error Correction. *Annals of the CIRP*. 1987; Vol. 36(No. 1): pp. 359-364.
2. Domnez, M.; Blomquist, R.; Hocken, R.; Liu, C.; Barash, M. General Methodology for Machine Tool Accuracy Enhancement by Error Compensation. *Precision Engineering*. 1986 Oct; Vol. 8(No. 4): pp. 187-196.
3. Furnival, G.M. (Yale University). Regressions by Leaps and Bounds. *Technometrics*. November 1974; Vol. 16(No. 4): pp. 499-511.
4. Kiridena, V. S. B.; Ferreira, P. M. (University of IL). Parameter Estimation and Model Verification of First Order Quasistatic Error Model for Three-Axis Machining Centers. *International Journal of Machine Tools and Manufacturing*. 1994; Vol. 34(No. 1): pp. 101-125.
5. Mallows, C.L. (Bell Laboratories). Some Comments on Cp. *Technometrics*. November 1973; Vol. 15(No. 4): pp. 661-675.
6. Soons, J. A.; Schellekens, P. H. (Eindhoven University of Technology). On the Calibration of Multi-Axis Machines Using Distance Measurements. *Proc. ISMQC*. 1992: pp. 321-340.
7. Soons, J.; Theuws, F.; Schellekens, P. (Eindhoven University of Technology). Modeling the Errors of Multi-Axis Machines: A General Methodology. *Precision Engineering*. 1992 Jan; Vol. 14(No. 1): pp. 5-19.
8. Teeuwssen, J.; Soons, J.; Schellekens, P. (Eindhoven University of Technology, Netherlands). A General Method for Error Description of CMM's Using Polynomial Fitting Procedures. *Annals of the CIRP*. 1989; Vol. 38(No. 1): pp. 505-510.
9. Zhang, G.; Chu, H.; Tang, W.; Jin, Z. (Tianjin University). Distance-Distance Method for Straightness Measurement. *Annals of the CIRP*. 1992; Vol. 41(No. 1): pp. 581-584.
10. Zhang, G.; Ouyang, R.; Lu, B.; Hocken, R. A Displacement Method for Machine Geometry Calibration. *Annals of the CIRP*. 1988; Vol. 37(No. 1): pp. 515-518.
11. Ziegert, J. C.; Mize, C. D. (Department of Mechanical Engineering, University of Florida). The Laser Ball Bar: A New instrument for Machine Tool Metrology. *Journal of the American Society for Precision Engineering*. 1994 Oct; Vol. 16(No. 4): pp. 259-267.

Numerical & Experimental Study of Ball-bar Dynamics during Probing

YuZhong Dai

Brown & Sharpe Manufacturing Company

North Kingstown, RI 02852

August 15, 1995

Abstract

This paper studies the dynamic response of a ball-bar during probing using nonlinear finite element modeling and experimental observations. The effects of probe type, contact direction, probing speed, probe pre/over-travel and ball-bar design on ball-bar dynamics are simulated. Major findings include: 1) Ball-bar deflection during probing is strongly dependent on the above mentioned factors; and 2) Higher probing speed reduces the ball-bar deflection at the moment of probe trigger; however, the consequently increased probe over-travel excites the ball-bar more severely and hence adds additional uncertainty to subsequent measurements if settling time between probeings is not adequate. Limited experimental observations agree favorably with the numerical simulation. All results presented in this paper are quantitative and may be readily extended to other test conditions.

1. Technical data of some probes

Relevant technical information on some commercially available probes are shown in Table I, where the Z is the axial direction of the stylus and X-Y is the plane perpendicular to Z.

Table I. Technical Data of Some Probes

Probe type	TP1	TP6	TP2-5W	TP7M
Stylus length(mm)	31	21	10	50
Delay time(μ s)	20	20	20	100-200
Pre-travel(μ m): X-Y max.	3.1 [†]	3.5 [†]	1.3 [†]	2.5 [‡]
Pre-travel(μ m): X-Y min.	1.8 [†]	2.3 [†]	0.7 [†]	2.5 [‡]
Pre-travel(μ m): Z	1.0 [†]	1.2 [†]	0.7 [†]	2.5 [‡]
Trigger force(N): X-Y max.	0.30 ⁺	0.25 ⁺	0.15 ⁺	0.05 [◆]
Trigger force(N): X-Y min.	0.15 ⁺	0.12 ⁺	0.07 ⁺	0.02 [◆]
Trigger force(N): Z	0.85	0.75	0.35	0.15-0.30 [◆]
Over-travel(mm)	8.5	6.5	4.5	5

[†] Tested at 8 mm/s speed. Results good for speed ranging from 1 mm/s to 15 mm/s.

[‡] Tested at 8 mm/s. More probing speed dependent due to longer response time.

⁺ Maximum/minimum force variation due to lobing effect of the kinematic arrangement.

[◆] Maximum/minimum force variation due to probe setting.

As can be seen from the table the probe force varies by as much as a factor of 15 along different contact directions and a factor of 7 among different probes. It varies by a factor of 2 even for the X-Y plane due to lobing effect. This makes ball-bar deflection dependent on probing practice.

2. The finite element model

ANSYS version 5.0 is used in simulating ball-bar dynamics during probing. Contact between the stylus tip of a probe and the ball of a ball-bar is simulated by a non-linear element COMBIN40.

The bar and the post are each modeled by 20 PIPE 16 elements equally distributed over their lengths. The post is rigidly fixed to the ground and is perpendicular to the bar. The bar and the post are made of steel. The finite element model is schematically shown in Fig. 1.

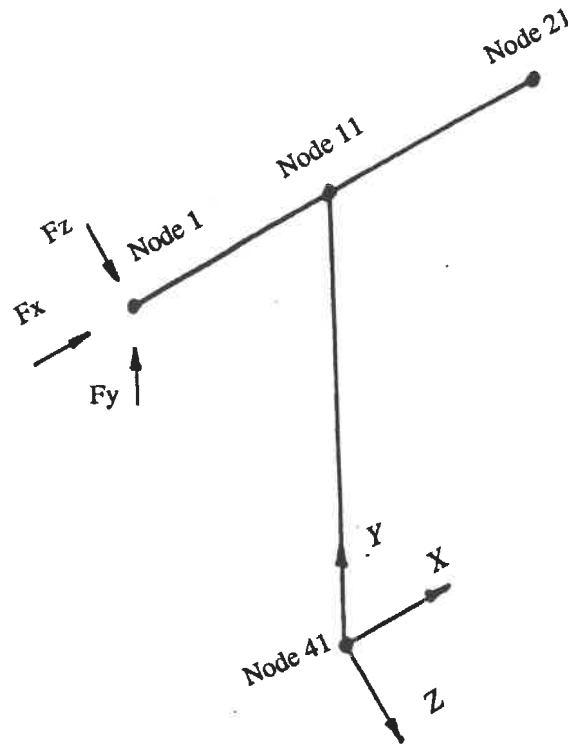


Figure 1 The finite element model

For quasi-static and transient studies load is applied at node one. The three dimensional displacement response at node one directly affects the ball-bar measurement and is therefore reported in this study.

Ball-bar designs with four different combinations of inner and outer diameters of the bar and the post, as shown in Table II, are investigated in this paper.

Table II. Ball-Bar Designs

Design	Bar OD	Bar ID	Post OD	Post ID
<i>A</i>	25	19	63	38
<i>B</i>	30	20	72	62
<i>C</i>	50	40	72	62
<i>D</i>	50	40	72	52

Note: All units in millimeters.

3. Ball-bar deflection under a static loading

Table III shows the deflection of the ball-bar under a static force of 0.5N at node one along X, Y, and Z directions, respectively.

Table III. Static Deflection of Ball-Bars

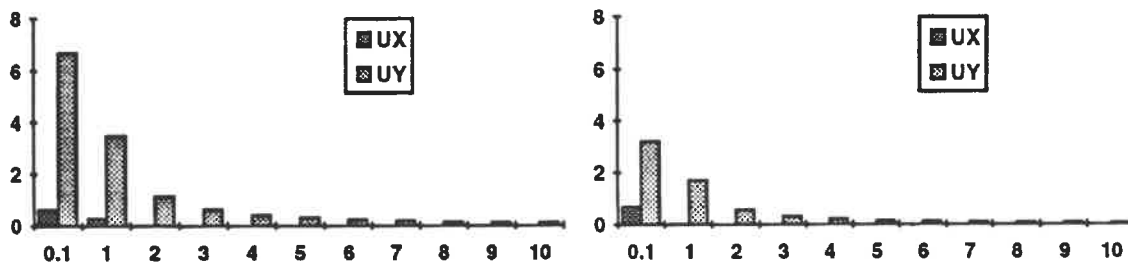
Design	Loading Direction	Component x	Component y	Component z
A	X	0.92	0.68	0
	Y	0.68	6.65	0
	Z	0	0	7.76
B	X	1.04	0.77	0
	Y	0.77	3.16	0
	Z	0	0	4.42
C	X	1.04	0.77	0
	Y	0.77	1.20	0
	Z	0	0	2.45
D	X	0.64	0.47	0
	Y	0.47	0.90	0
	Z	0	0	1.68

Note: All units in micrometers.

This shows the strong dependence of ball-bar stiffness on contact direction. Taking design *B* as an example, the stiffness along X direction is 8.5 times of that along Z direction. Design *C*, on the other hand, is not only much stiffer but also well balanced.

4. Effect of probing speed on ball-bar deflection

Simulation results showing the dependence of ball-bar deflection on speed at the moment of probe trigger is shown in Fig. 2. The horizontal axes are probing speed in *mm/s* and the vertical axes are deflection in X and Y direction in micrometers. As can be seen, ball-bar deflection at the moment of probe trigger may differ significantly when the probing speed varies. For example the deflection reduces by a factor of ten when speed increases from 1 to 5 *mm/s*(Design A).



(a) Contact along Y axis(Design A)

(b) Contact along Y axis(Design B)

Figure 2 Effect of probing speed on ball-bar deflection

5. Impact of probe over-travel on ball-bar dynamics

Although the deflection of a ball-bar caused by probe over-travel does not have any effect on an initial probing, it may excite the ball-bar and add additional uncertainty to a subsequent measurement if settling time between probeings is not sufficient. Tests were conducted to observe the dynamic response of a ball-bar during probing along the Z direction of the ball-bar. Probing was done by a TP2-5W probe along its axial direction. The trigger force was about 0.3*N*. A laser interferometer was used to measure the deflection of the ball-bar. Figure 3, (b) is a subset of (a), shows some test results of ball-bar deflection(μm) versus time(*ms*).

Contact speed: 7.6mm/s; Time interval of data acquisition: 1 ms.

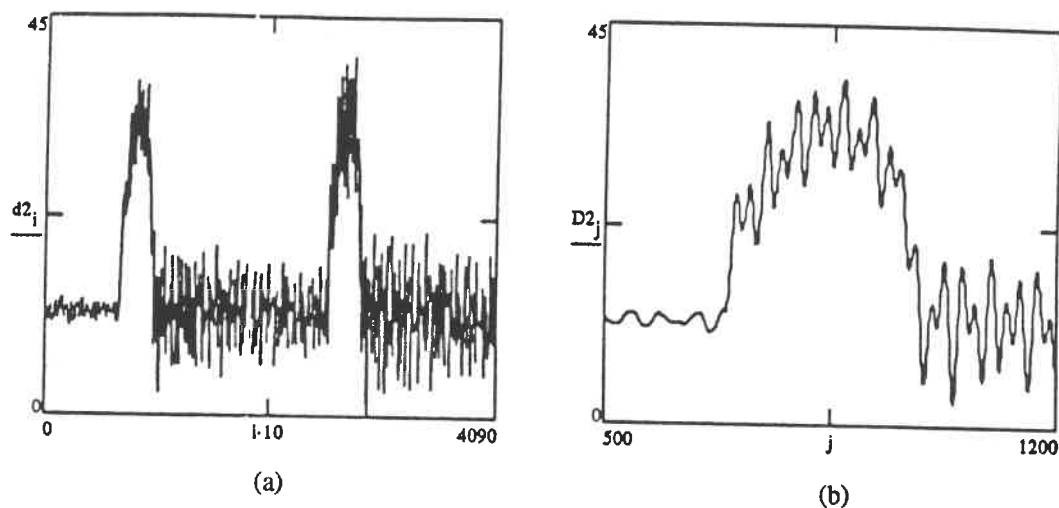


Figure 3 Dynamic response(experimental) during probing for ball-bar design B

6. Conclusion and application to probing strategy

- 1). Ball-bar stiffness: Ball-bar's stiffness is strongly direction dependent. The axial direction of the bar is the stiffest while the twist about the post axis is the weakest.
- 2). Probe force variation: Probe force is also direction dependent even for a given probe under a certain setting condition. When ball-bar deflection is significant, probing along the axial direction of a probe stylus should be avoided or used only along the stiffest direction of the ball-bar.
- 3). Proper probing speed: Slow speed results in excessive ball-bar deflection. Higher speed, on the other hand, results in more over-travel. The latter will excite the ball-bar structure more severely and hence add additional uncertainty to subsequent measurements.
- 4). Probing strategy: The combination of probe orientation, probing speed, and contact direction can be carefully chosen to reduce variation in deflection of a ball-bar at one location. Similar practices can be used for different ball-bar orientations to improve measuring accuracy.
- 5). Ball-bar design: A better design with proper geometry and damping may significantly reduce measurement uncertainty due to static and/or dynamic deflection of the ball-bar.
- 6). Software correction: It may be introduced to reduce the impact of the direction dependence of both ball-bar stiffness and probe force on measuring accuracy of a ball-bar.

Acknowledgments

Guidance from Thomas Charlton Jr., assistance of Paul J. Anderson in setting up the test apparatus, and help from C.S. Reid of Renishaw plc. are gratefully acknowledged.

References

- [1] "Static and Dynamic Properties of Free Standing Ball Bars", by S.D. Phillips et al. of NIST and J. Henry of Sheffield Measurement Corporation. *Proceedings of ASPE Conference, 1991.*
- [2] Letters to the author from C.S. Reid, Product Marketing Engineer of Renishaw plc., on May 12th and 23rd, 1994.

Automatic measurement of axial and tilt error motions of an aerostatic spindle using optical interferometry.

A Idowu and A E Gee
School of Industrial and Manufacturing Science
Cranfield University

ABSTRACT

There is general need in the field of precision engineering for spindles of machine tools and measuring systems to perform to sub-micrometre accuracies. In order to verify the dynamic performance of the spindle, "parasitic error" motions (i.e. unwanted tilts, radial and the axial motions in any of the five degrees of freedom) must be precisely measured. An optical interferometric approach is used, where-by interferograms from the rotating artefact are acquired, digitised, processed and analysed in order to extract the required measurement of the spindle's tilt and axial error motions.

1. INTRODUCTION

Earlier work at Cranfield has shown the potential application of optical interferometry for measuring the performance of a research machine tool aerostatic spindle [1]. Other possible techniques include arrays of closely-mounted proximity sensors to measure the run-out of an artefact, using capacitive [2] or backscatter optical fibre [3] sensors.

The current work utilizes a Fizeau interferometer which, for development, is mounted on a vertical spindle diamond turning machine tool as shown in figure 1. Interferograms from the rotating artefact mounted on the spindle face-plate, are acquired and digitised with the aid of an 8-bit monochrome frame grabber which is adequate for this application. The interferogram data is sampled at selected angular positions of the spindle-rotation and image processing techniques are used to filter and modify the stored fringe pattern, to enable the measurement of the spindle tilt and axial displacement.

2. INTERFEROMETER ADJUSTMENT AND OPERATION

In order to measure spindle run-out, it is necessary to adjust the tilt of the interferometer reference surface normal to be close to parallel with the spindle axis. This is achieved by using two on-off servomechanisms driving screws which tilt the reference surface about two orthogonal axes lying in a plane normal to the optical axis. As shown in figure 2, reference surface tilt-adjustments are made at points A (about axis OB) and B (about axis OA).

When interferometrically examining a stationary surface, fringe spacing and orientation are controlled respectively by the angle between the normals to the reference and test surfaces and by the pointing direction of the tilt of the plane containing them, both of which are held constant for a measurement. Unlike this, in the case in hand, with the

test-piece rotating about an axis not necessarily parallel to the reference surface normal, the fringe dynamics are not straightforward. With a plane test artefact (normal **A**) rotating on a perfect spindle (axis **S**) monitored by an interferometer (reference surface normal **I**), these three axes shown in figure 3 must be considered.

In general with respect to the spindle axis, the inclination of the interferometer axis ϕ_I will not equal that of the artefact ϕ_A . The orientations of the reference and test wavefronts can be described by the position of their point-spread functions in the focal plane, with respect to the axis of rotation. The loci of the test **A** and reference **I** wavefront normals [4], prescribe the motion of the point-spread functions with respect to **S** the spindle axis, that of **A** prescribes a circle while that of **I** remains stationary.

Figures 4a - 4d show a plan view of the focal plane for different recognisable conditions, where vector **SA** represents the tilt of the test surface normal with respect to the axis of rotation **S**, vector **SI** gives the tilt of interferometer reference axis **I** with respect to **S**, and **IA** gives the resultant magnitude and direction of the tilt between the test and the reference axes and defines the number and orientation of the fringes observed.

The following conditions can be observed in the focal plane of the interferometric system [1]:

If $\phi_A > \phi_I$ the fringe pattern will rotate with the fringe spacing varying cyclically (see figure 4a).

If $\phi_I > \phi_A$ the fringe pattern will oscillate through partial rotations and the spacing will vary, (see figure 4b).

Special cases observed are:

When $\phi_A = 0$, the fringe pattern will remain stationary (as the spindle rotates).

When $\phi_I = \phi_A$, at one particular point in each revolution, the fringes will 'fluff-out' to infinity (see figure 4c).

Obviously with $\phi_I = \phi_A = 0$, the interferometer will be in 'normal' adjustment and aligned with the spindle axis (yielding infinite fringe spacing).

When $\phi_I = 0$ and $\phi_A = 0$, fringes of constant spacing will be observed rotating at a constant rate (as shown in figure 4d). This is the condition required for the spindle metrology system.

3. INTERFEROGRAM ANALYSIS

After pre-processing (i.e. spatial filtering, recursive averaging, and background subtraction) and processing (adaptive thresholding, skeletonisation and thinning), the interferogram is analysed to identify the positions of maximum and minimum fringe spacing, which occur when the spindle axis **S**, the test-piece normal **A** and the reference surface normal **I** lie in the same plane. In this condition, the tilt of the interferometer (i.e. of the reference surface normal with respect to the spindle axis **S**) is adjusted (using the servomechanisms) to equalise the fringe tilt for the full revolution so that fringes of constant spacing are observed throughout the spindle rotation.

Once this condition has been achieved, spurious error motions can be detected with the aid of image processing techniques. Detected changes in phase of the fringe pattern will be a measure of axial error motion; changes in fringe spacing will be a measure of spurious tilts with respect to the spindle axis. To observe radial error motions of the spindle a modified optical arrangement using an aplanar artefact (e.g. a spheric) would be required.

4. REFERENCES

1. Gee, A.E , McCandlish, S.G & Puttick, K.E "Interferometric monitoring of spindle and workpiece on an ultra-precision single-point diamond facing machine" Proc. SPIE 1988, Vol 1015, pp74-80.
2. Chapman, P.D "A capacitance based ultra-precision spindle error analyser" Precision Engineering 1985, Vol7, pp 129-137.
3. Muzuno, H "Micro-Tilt controlled rotating face-plate for single-point diamond turning" PhD Thesis 1993, Cranfield University, pp17-22, pp98-99.
4. Gee, A.E "Dynamic rotational interferometry" in Blakshaw, D.M.S et al Laser Metrology and Machine Performance Computational mechanics Publications Southampton / Boston 1993, pp203-210.

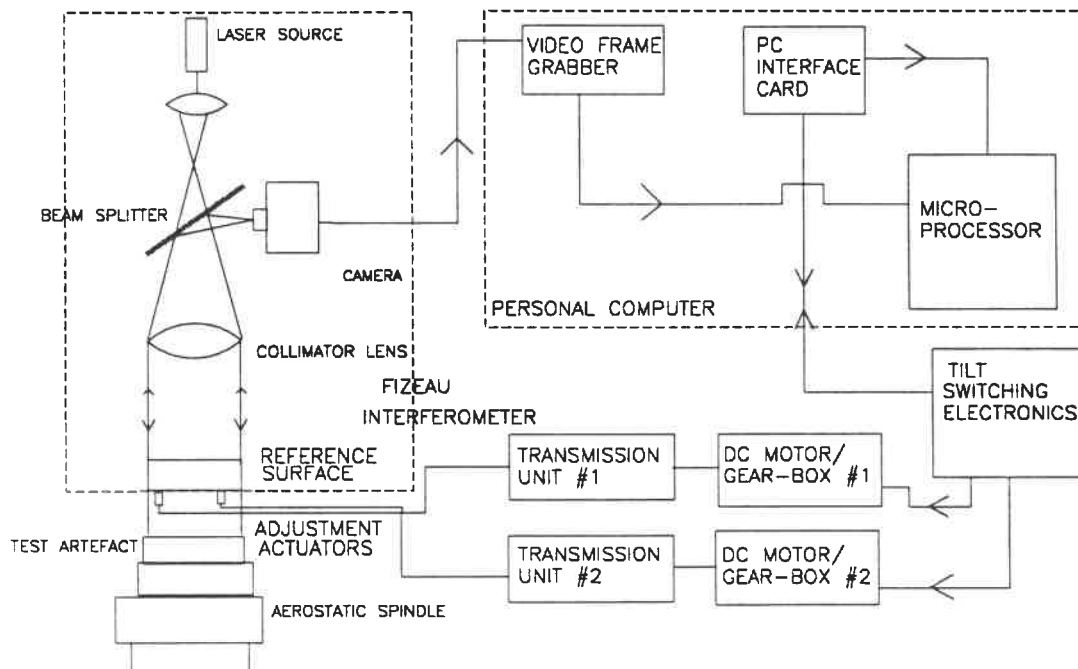


FIGURE 1

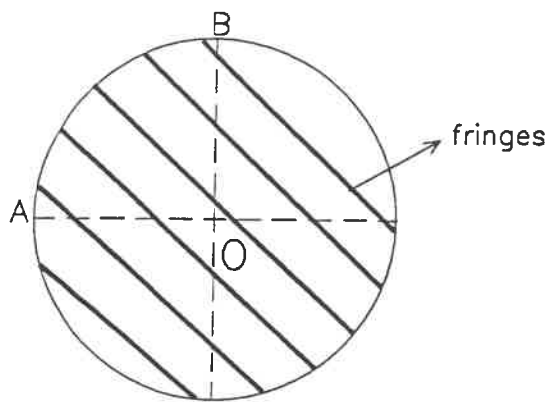


FIGURE 2

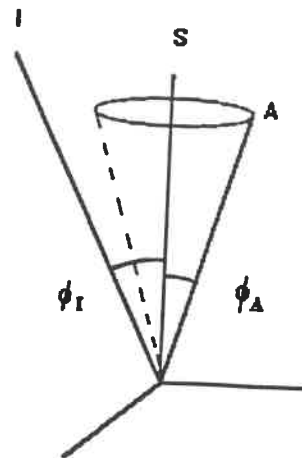


FIGURE 3

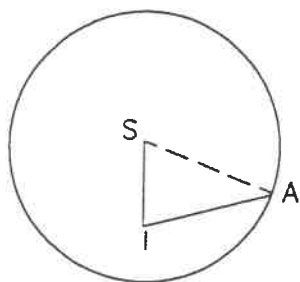


FIGURE 4a

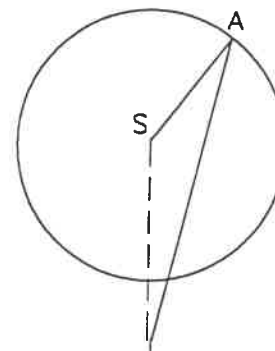


FIGURE 4b

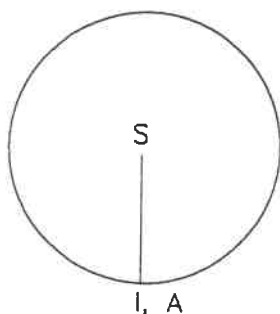


FIGURE 4c

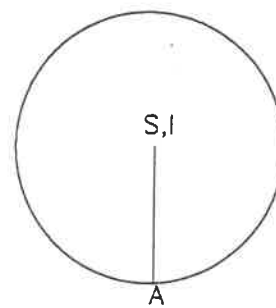


FIGURE 4d

On-Machine Acceptance of Machined Components

Andrew J. Hazelton
Sandia National Laboratories
Livermore, California

Abstract

A procedure for accepting small lots of machined parts on the machine tool with no need for post-process inspection has been developed. In this procedure, a positional error model of the machine is first developed. This model is used to determine the uncertainty of the machine position based on the variances of the individual error elements. The positional uncertainty can be extended to cover touch probe measurement uncertainty. This approach was tested on a set of turned parts. The results show that this method can be used to estimate the uncertainty of an on-machine inspection process and to certify this process.

Notation

aT_b	HTM describing position and orientation of coordinate system b in coordinate system a
x, z	position of x and z axes
X_c, Z_c	tool length offset
X_w, Z_w	work offset
$\delta x, \delta z$	total error in X and Z
$\delta x(a), \delta z(a)$	translational error in X and Z due to axis a
$\epsilon y(a)$	yaw error due to axis a
$\alpha_{par}, \alpha_{squ}$	axis alignment error
μ_1, μ_0	upper and lower bounds of uniform distribution

Introduction

The streamlining of the nuclear weapons complex has brought on the need for quicker manufacturing response to smaller lot sizes. The strict quality and certification requirements, however, have not changed. In order to produce the smaller lots of parts for the same unit cost and high certification levels, a new approach to small lot quality control is required.

In traditional large lot manufacturing, statistical process control (SPC) techniques are often used to certify part quality. Several techniques have been attempted to apply SPC techniques to small lot processes. A common approach is to normalize the attribute data by subtracting the nominal value of the measurement.¹ Another technique is to compute a process capability index² for the machine's ability to make a class of features. Both methods require a portion of the production run to establish the limits of control, and each method ignores the actual cause of product variability.

Another approach to this problem is to inspect the part on the machine. Although this has been applied successfully in some cases,³ the method is

criticized because errors made in machining the part due to machine geometry errors are repeated, one-for-one, when inspecting the part.

One alternative is to certify the process by certifying the basic accuracy of the machine. This approach is based on the deterministic manufacturing principle: "machine tools obey cause and effect relationships that are within our ability to understand and control and there is nothing random or probabilistic about their behavior."⁴ Using this approach, the process engineer assesses the accuracy of individual elements of the machine tool, then combines the accuracies to determine the machine's ability to produce a good part.

The traditional method for combining these error terms is the error budget. With this method, individual machine error components are identified, estimates or measurements of their magnitude are made, and a suitable combination rule is applied to determine the total machine error.⁵ This method has been applied successfully to precision machine design for a number of years,⁶ but has not been verified for process certification.

The recent adoption of the concept of measurement uncertainty by international standards bodies⁷ has provided a framework for understanding the accuracy of measurement results and comparing the relative quality of separate measurements of the same part.

The purpose of this work is to develop a procedure for accepting small lots of machined parts on the machine tool with no need for post-process inspection. This capability will be achieved when the uncertainty of the on-machine inspection is sufficiently low to guarantee the part meets its tolerance requirements. To do this, a homogeneous transformation matrix (HTM) of the machine tool will be used to generate a positional error model. This model will then be used to generate a kinematically correct error budget of the machine tool. Statistical rules will then be used to

generate an estimate of the measurement uncertainty of the machine for different features. Finally, the measurement uncertainty will be tested against actual part data for accuracy.

HTM model of the machine

The components in this work are stainless steel gas bottles machined on a LeBlond Baron 25 slant-bed lathe. A schematic of the machine with a coordinate frame attached to both the workpiece and the tool is shown in Figure 1.

The lathe consists of seven major independent elements: the machine base, the z-axis slide, the x-axis slide, the turret, the cutting tool, the spindle, and the workpiece. A coordinate frame is attached to each of these elements. Considering the relationships of the geometry of the machine tool, an HTM can be developed describing the position of an element in its neighbor's coordinate system.⁸ Consecutive HTMs can be multiplied to describe the position of any element within any other element's coordinate frame. For instance, the position of the cutting tool in the coordinate system of the workpiece can be described by:

$${}^wT_c = {}^wT_s {}^sT_b {}^bT_z {}^zT_x {}^xT_t {}^tT_c \quad (1)$$

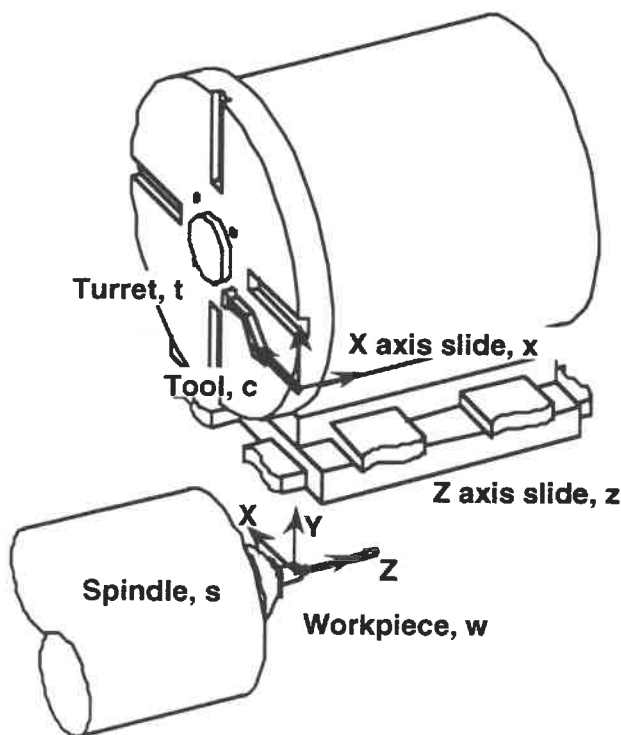


Figure 1 Schematic of the LeBlond lathe

Assuming perfect geometry of the machine tool, the resulting transformation is:

$${}^wT_c = \begin{bmatrix} 1 & 0 & 0 & -X_c + x + X_w \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & -Z_c + z + Z_w \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (2)$$

All real machine elements, however, have errors in their motion and alignment. Assuming the elements are rigid bodies, each element has six error motions, three translational and three rotational. Because the LeBlond is a 2-axis lathe, many of the error motions are in the non-sensitive direction and can be ignored. From the resulting transformation, the error term in X is:

$$\begin{aligned} \delta x = & \delta x(x) + \delta x(z) + \delta x(t) + \delta x(c) - \delta x(s) \\ & - \delta x(w) - (\epsilon y(x) + \epsilon y(t) + \epsilon y(z) - \epsilon y(s))Z_c \\ & - (\epsilon y(s) - \alpha_{par})z \end{aligned} \quad (3)$$

A similar expression can be developed for the error in Z. These terms describe the error of the position of the tool with respect to the workpiece in the workpiece coordinate system. If the error terms are known as a function of the position of each machine element, this equation could be used to correct the position error of the machine tool. In this case, the errors are assumed to be known only as the maximum and minimum values in the work volume.

Error budget

The error equations can be used to develop an error budget for the inspection process that is based on the kinematics of the machine. For a linear function of several uncorrelated variables, $h[x_1, x_2, \dots, x_n]$, the expected value is:⁹

$$E(h[x_i]) = h[E(x_1), E(x_2), \dots, E(x_n)] \quad (4)$$

and the variance is:

$$Var(h[x_i]) = \sum_{i=1}^n \left(\frac{\partial h}{\partial x_i} \right)_{E(x_i)}^2 Var(x_i) \quad (5)$$

The expected value of the error in X is merely (3) evaluated at the expected value of each term. The variance of the error in X is:

$$\begin{aligned} Var(\delta x) = & Var(\delta x(x)) + Var(\delta x(z)) + Var(\delta x(t)) \\ & + Var(\delta x(c)) + Var(\delta x(s)) + Var(\delta x(w)) \\ & + (Z_c)^2 (Var(\epsilon y(x)) + Var(\epsilon y(t)) + Var(\epsilon y(z))) \\ & + (Z_c - z)^2 Var(\epsilon y(s)) + z^2 Var(\alpha_{par}) \end{aligned} \quad (6)$$

Similar expressions can be developed for $E(\delta z)$ and $Var(\delta z)$.

Over the work volume of the machine, each error term can be considered to be uniformly distributed,⁵ with expected value:⁹

$$E(u) = \frac{\mu_0 + \mu_1}{2} \quad (7)$$

and variance:

$$Var(u) = \frac{(\mu_1 - \mu_0)^2}{12} \quad (8)$$

Machine data can be used to calculate the expected value and variance of each error term. The variance of the total error can be determined with (6). The resulting distribution is assumed normal by application of the Central Limit Theorem.

Application to features

The above expressions describe the error and variance of single probed points. But features are measured using combinations of these probed points, thus alternate expressions must be developed. The mean diameter of a cylinder can be expressed as the average of several measurements in the X direction, or:

$$\bar{\phi} = \frac{1}{N} \sum_{i=1}^N X_i \quad (9)$$

The variance of the diameter measurement can be expressed as:

$$Var(\bar{\phi}) = \frac{1}{N} Var(\delta x) \quad (10)$$

Cylindricity can be expressed as:

$$Cyl = \max(X_i) - \min(X_i) \quad (11)$$

A conservative estimate of the variance is:

$$Var(Cyl) = 2 \cdot Var(\delta x) \quad (12)$$

Using these expressions it is possible to express the uncertainty of measurements made with an on-machine inspection process. To do this the user first determines the level of confidence required of the measurement. For large N, a table of the standard normal distribution can then be used to translate the confidence level into a number, n_p , of standard deviations. Because the standard deviation is simply the square root of the variance, the uncertainty of the measurement at the given confidence level can be expressed as:

$$U_p(\bar{\phi}) = \pm n_p \sqrt{Var(\bar{\phi})} \quad (13)$$

Experiment

A set of ten test parts were machined to test the on-machine acceptance method.

Machine tool metrology

First the geometry of the lathe was measured using standard techniques⁸ to determine the values and distributions of the error terms. A working volume of approximately 7 inches in Z and 3 inches in diameter (X) was chosen. The difference between the maximum and minimum values in this volume of each error term was recorded (Figure 2). With the exception of the squareness and parallelism errors, each term was considered to be centered around zero. The parameters μ_1 and μ_0 were calculated as plus and minus half the total spread. Finally the expected value and the variance were determined using (7) and (8). It should be noted that the spread determined above is the same parameter that is measured when a machine is tested in accordance with the ANSI/ASME B5¹⁰ standards.

Several of the parameters, including spindle error motions and thermal components of rigid body errors, are estimated based on data from other machines or information provided by manufacturers.

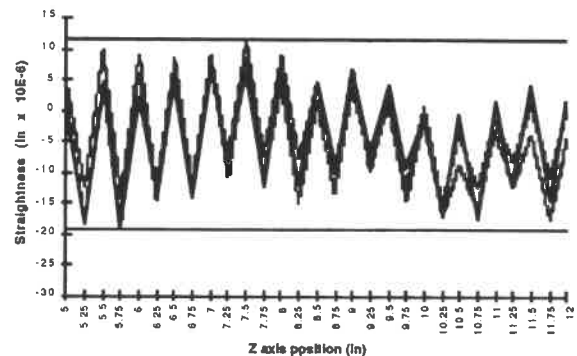


Figure 2 Z axis straightness with μ_1 and μ_0 shown

Error Budget calculation

Using (6), the kinematic error budget can be calculated based on the individual variances. For reference purposes a traditional error budget for the machine was calculated. Assuming the standard rules of thumb produce a ± 2 sigma estimate,¹¹ the results are compared in Table 1.

Based on the metrology data, $Var(\delta x)$ is estimated as 1.88×10^{-6} . Using (10), (12), and (13), the uncertainty of diameter to the 99%

confidence level is $\pm 772 \times 10^{-6}$ in based on a 21 point measurement. The estimated uncertainty of cylindricity is 5.00×10^{-3} in.

Error Budget Method	Error Estimate (in)
$0.6 \times \sum \text{Arith}$	0.005905
$\frac{\sum \text{RMS} + \sum \text{Arith}}{2}$	0.006408
$\pm 2 \times \sqrt{\text{Var}(\delta x)}$	0.005485

Table 1 Error budget comparison

Results

Ten parts were machined and inspected on the lathe. These parts were then taken to a separate CMM where they were inspected. The results are listed in the table, with the 99% uncertainty band calculated as in (13).

Feature	Average difference (CMM-OMI) (in)	99% uncertainty band (in)	Percentage of measurements falling into band
0.4330" $\varnothing$	475×10^{-6}	772×10^{-6}	60%
1.5000" $\varnothing$	525×10^{-6}	754×10^{-6}	50%
Cylindricity	110×10^{-6}	5.00×10^{-3}	100%

Table 2 Experimental results

The results show a significant systematic difference between diameters measured on the machine and independently. In addition, the true value of the diameter is at the edge or outside the uncertainty band. This is explained by considering the accuracy of diameter measurements depends on the absolute accuracy of the machine in X relative to a datum fixed in space, the centerline of the spindle. The assumption of zero expected value for error terms in X ignores the absolute error and is most likely too liberal an assumption.

There is a smaller systematic difference between cylindricity measurements. The true value of the measurement is well within the uncertainty band. This is because cylindricity is a relative measurement between a set of points and absolute accuracy of the points is unimportant.

These results suggest more work is necessary to ascertain the absolute uncertainty of diameter measurements on the machine. For relative

measurements, however, the uncertainty model appears to be satisfactory.

Conclusions

A method for determining the uncertainty of the on-machine inspection process was presented. This method utilizes an HTM model of the machine tool to develop a kinematically correct error budget. Basic assumptions on the nature and distribution of individual machine error terms can be used to assess the positional error of the machine. From the position error budget, a feature error budget can be derived. Standard measures of machine tool accuracy, such as those generated when a B5 test of the machine is made can be used to numerically assess the errors. Finally, experimental results show that actual measurements fall within measurement uncertainty of the true values.

In the future this work will be extended to test the validity of statistical assumptions regarding error terms, and extended to cover a wider range of features.

Acknowledgments

This work was supported by the Department of Energy under contract DE-AC04-94-AL85000. In addition, the author would like to thank Debra Krulewich and Charlie DeCarli for their insightful comments.

References

- Bothe, D. R. "A powerful new control chart for job shops," *ASQC Quality Congress Transactions*, Toronto, 1989, 265-270.
- Gitlow, H. S., et al. *Quality Management: Tools and Methods for Improvement*, 2nd ed., Richard D. Irwin, Inc., 1995.
- Taylor, B. R. "Dimensional control through the tolerancing of the solid model," *Proceedings of the 1993 International Forum on Dimensional Tolerancing and Metrology*, Dearborn, MI, 1993, 161-166.
- Bryan, J. B. "The deterministic approach in metrology and manufacturing," *Proceedings of the 1993 International Forum on Dimensional Tolerancing and Metrology*, Dearborn, MI, 1993, 85-96.
- Donaldson, R. R., "Error Budgets," *Technology of Machine Tools*, v.5, 1980, sec. 9.14, 1-14.
- Thompson, D. C. "The design of an ultra-precision CNC measuring machine," *CIRP Annals*, 1989.
- Phillips, S. D., et al. "Measurement uncertainty considerations for coordinate measuring machines," NISTIR 5170, 1993.
- Slocum, A. H. *Precision Machine Design*, Prentice-Hall, Inc., 1992.
- Hahn G. J. and S. S. Shapiro. *Statistical Models in Engineering*, John Wiley & Sons, Inc. 1994.
- ANSI/ASME B5.54. *Methods for Performance Evaluation of Computer Numerically Controlled Machining Centers*, American Society of Mechanical Engineers, 1993.
- Shen Y. L. and N. A. Duffie. "Comparison of combinatorial rules for machine error budgets," *CIRP Annals*, 1993, 619-622.

Correlation Models for Improved Machine Tool Accuracy

**Nandakumar Srinivasan, Fred Farabaugh,
Robert G. Wilhelm and Robert J. Hocken
The University of North Carolina at Charlotte
Charlotte, NC 28223**

Prediction of variations in the geometry of a part produced by a specific machine tool can contribute significantly to accuracy improvement in precision manufacturing processes. Several approaches have been used by precision engineers to improve the performance of machine tools using mathematical models and parametric error measurements [1, 2]. The models derived in these approaches, in most cases, require extensive instrumentation, testing and measurement, requiring significant investment in time and resources. Moreover, such models may not be accurate over a period of time and, hence, the machine must be tested and re-characterized regularly. Our research focuses on development of fast and cost effective methods for improving machine tool accuracy with the aid of simple and standardized machine tool performance tests.

An adaptive error prediction method using feature-based analysis techniques is under development for improved accuracy of computer numerically controlled (CNC) machining centers and manufacturing processes. This technique correlates dimensional and form errors of features in manufactured parts to systematic machine tool errors. Standard performance tests are used as specified in the ANSI/ASME B5.54 standard- "Methods for Performance Evaluation of Computer Numerically Controlled Machining Centers" [3]. The advantage of this approach is that these performance tests can be completed in a few hours while other approaches may require days or weeks of parametric error measurements. If a standard test condition is specified, comparative assessment of similar machines can be made.

Machining centers are designed to accept a wide variety of parts of different sizes and geometries. Hence, no one part can be used to determine whether a machine will produce all types of parts within required tolerances. However, simple test parts, incorporating common features, can be designed to characterize machine performance for features common to a wide variety of parts. The machined accuracy of common features is then used to represent accuracy of typical production parts. In this research, at the initial stage, only a limited number of features and operations are considered. Tool dominated operations, such as tapping, are specifically excluded. A set of diagnostic rules that relate machine errors with part errors have been derived for analyzing the effect of error components on part features. The diagnostic rules help to identify simple parts that correlate one or two machine errors with one part feature [5]. Working from the knowledge of errors in a particular machine tool, three types of test parts are designed to isolate the error effects caused by one or a combination of error components of the machine. The three types of test parts [5] are test parts with prismatic features, test parts with prismatic and cylindrical features and test parts with contour features as shown in Figure 1.

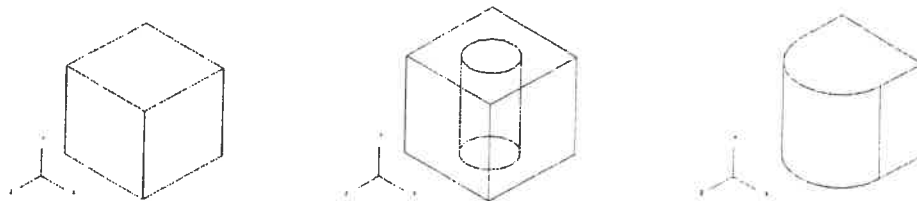


Figure 1. Three Types of Test Parts

Using these test parts, an adaptive algorithm, shown in Figure 2, is under development . A CAD model of the part to be manufactured is developed. The part is then machined using a B5.54-tested CNC machining center and measured using a software-corrected coordinate measuring machine. Repeated machine tool performance measurements, cutting tests, and measurements of the part are done to collect the data used for deriving the correlation. The data obtained from measurements is compared with the CAD model and variations in accuracy of the test part are determined. The results obtained from the standard tests specified in the ANSI/ASME B5.54 Standard for performance evaluation of CNC machining centers should predict variations of the machine tool. The variations of the machine tool and variations of the test part are used to derive a predictor that correlates variations in the measured geometry with variations in machine's performance. The results are then fed back to the CAD model to increase the accuracy of the part produced by altering the design of the test part.

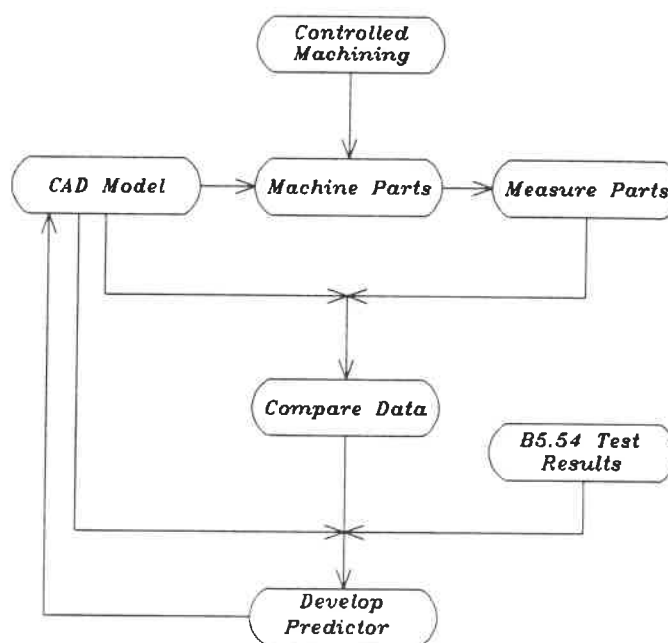


Figure 2. Adaptive Correlation Algorithm

A series of test parts are machined and inspected to establish correlation models. Machining accuracy is governed by different complex interactions of several effects. Due to the many possible variations in cutting conditions, it is not practical to check the

performance of a machine tool based on machining tests. However, machining tests would result in some information that can be used to derive correlation between measured machine and part errors. Simple test parts are designed using diagnostic rules that relate machine errors with part errors. A list of machine errors that affect part errors in all three types of parts is developed. Donmez has developed a similar set of diagnostic rules, in the form of mathematical equations, to relate errors of a turning center with the parts produced by them [4]. Deriving diagnostic rules for a machining center is more complicated for a machining center due to more axes and errors related to combinations of axes. For the machining center, instead of mathematical equations, diagnostic rules are stated as interaction matrices that relate machine errors and part errors. This helps to qualitatively determine which features should be present in test parts and also to estimate the magnitude of errors in part features for validation purposes. For instance, flatness of a surface is primarily dominated by roll and yaw of the machine table and compliance. If the roll and yaw of the machine table are 7.5 arcsec and 6.25 arcsec respectively, the combination of the angular errors would induce a flatness of 19.1 μm for a part of length 279.4 mm and width 203.2 mm. The flatness over the same zone measured from a machined test part was 18.9 μm . Similarly, the predicted straightness error in the part of 7 micrometers closely associates with the measured straightness error. Hence, diagnostic rules are used to validate the measured errors in part features in addition to aiding in the design of test parts.

The error sources that affect dimensional and form errors in parts can be classified into three stages - Process Planning, Machining, and Post-Process Inspection [6]. In the process planning stage, effects of errors can be avoided or reduced by careful NC programming, fixturing, and alignment techniques. Fixturing errors can be significant, especially in horizontal spindle machining centers where the part is mounted to an angle plate while machining. Thermal expansions and contractions cause distortions in the part as fixturing constrains the part from expanding freely due to thermal effects. Several experiments were conducted to obtain a fixturing method that reduces the effects of fixturing on part errors. Three identical precision-ground steel washers were sandwiched between the angle plate and test part to eliminate large mating surfaces between test part and angle plate, minimizing area of contact and also establishing a three-area contact. Standard torque charts were used to uniformly torque the fixturing bolts thereby preventing bow in the part due to excessive torque. Fixturing effects due to the angle plate and changes in temperature were determined by measuring the part with and without the angle plate at ambient temperature (25° C) and at 20° C. It was clearly evident that the temperature change did not induce significant effect on flatness on part surface but removing the part from angle plate significantly influence flatness measurements. To minimize temperature and fixture effects, the test parts are machined and measured on the angle plate. Other possible error sources like tool wear, workpiece location, workpiece deflection are minimized by careful process planning. In the machining stage, part errors are primarily influenced by mechanical loading and thermal effects generated by cutting loads, machine operations, and coolant flow over part and machine surfaces. Temperature drifts in the shop were monitored to observe any dramatic changes in ambient temperature. Experiments were conducted to observe effects of coolant flow by

monitoring temperature of coolant at different locations with three different load cycles which included actual machining of the part. It was determined that the major source of heating of coolant was the coolant pump and not the machining operation and the temperature rise of the coolant was less than 0.7°C . The errors in the measurement stage were reduced by careful coding of automatic measurement routines for the CMM. Experiments to test repeatability of the probe and repeatability of measured data were conducted to validate the measurements performed.

It has been observed that process errors contribute significantly to errors in the part. Hence, in order to correlate machine errors with part errors, it is desired to isolate the effects of process errors or to control them. A series of test parts are machined using design of experiments techniques (2-level, three factor and four factor fractional factorial methods) to determine the effects of process variables on accuracy of part features. The experimental designs are formulated to analyze the impact of extraneous variables like thermal effects, feeds, spindle speeds, tooling and the like, on part errors and these effects are then minimized by controlling these variables suitably. In order to isolate the effects of machining process on part errors, the machining centers used to cut the test parts are software-corrected off-line for its geometric errors. The resulting errors on the part are due to the process only. Machining experiments are conducted with and without software-correcting the machine.

The predictor developed can be used for many different parts since most production parts can be decomposed into a set of basic features. The predictor could be used to determine whether a part, composed of a set of basic features, could be manufactured by a particular machine within the specified tolerances. This would considerably reduce the time required for traditional post-process quality control operations and the amount of scrap material, while increasing the production rate. As well, since the predictor relates machine errors to part features, the correlation matrix could be used to guide designers in the selection of tolerances. In this way designs might be developed to avoid the dominant errors of the machine tool.

1. M.A. Donmez, "Development of a New Quality Control Strategy for Automated Manufacturing", Proceedings of Manufacturing International 1992, D. Durham, ed., ASME, NY, 1992.
2. J.G. Salsbury, "Development of a Methodology correlating machine tool errors to work piece geometry", M.S. Thesis, University of North Carolina at Charlotte, 1991.
3. ASME B5.54-1992, "Methods for Performance Evaluation of Computer Numerically Controlled Machining Centers", The American Society of Mechanical Engineers, 1992.
4. J. Mou, M.A. Donmez and S. Cetinkunt, "An Adaptive Error Correction Method using Feature-based Analysis Techniques for Machine Performance Improvement - Part I : Theory Derivation", Accepted for publication in ASME Journal of Engineering for the Industry.
5. N. Srinivasan, F. Farabaugh, R.G. Wilhelm and R.J. Hocken, "Correlating Machine Tool Accuracy with Part Features", Proceedings of ASPE 9th Annual Meeting, October, 1994.
6. Kiriden, V.S.B., Ferreira, P.M., "Kinematic Modeling of Quasistatic Errors of Three-Axis Machining Centers". Int. J. Mach. Tools Manufact., Vol. 34, No. 1, 1994.

Error Compensation for CMM Touch Trigger Probes

W.Tyler Estler, S.D. Phillips, B. Borchardt, T. Hopp, C. Witzgall
M. Levenson, K. Eberhardt, and M. McClain

National Institute of Standards and Technology
and

Y. Shen and X. Zhang
George Washington University

We present the analysis of a mechanical model of a touch trigger probe. Such probes are widely used on modern coordinate measuring machines (CMMs). The most common design employs a kinematic seat arrangement that simultaneously provides the switching mechanism upon part contact and the probe relocation and overtravel protection functions [1-4]. A sketch of such a probe is shown in Fig. 1.

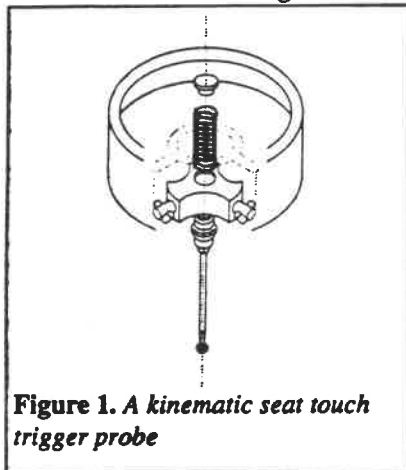


Figure 1. A kinematic seat touch trigger probe

The stylus is attached to a tripod structure whose three cylindrical legs are kinematically located by three pairs of crossed cylinders. A spring, coaxial with the stylus, supplies a seating preload. The trigger signal for this type of probe is an increase in the series electrical resistance through the contacting elements, above a preset limit, caused by displacement of one or more of the tripod legs as the stylus is deflected during part contact.

Motion of the CMM between the positions of part contact and probe trigger is called *pre-travel*.

The existence of pre-travel, by itself, is not a source of measurement error since probe calibration procedures, using a master sphere of known radius, determine an effective stylus ball radius that compensates for the average pre-travel. Of much greater significance for high-accuracy applications is the direction-dependent *pre-travel variation* or *probe lobing*, so-called because the pattern of variation often displays lobes that reflect the three-fold symmetry of the probe construction. The presence of probe lobing can be a dominant source of error in highly repeatable machines and may cause significant errors in many CMM measurements [5,6].

The characteristic probe lobing pattern is well illustrated by Figure 2 which shows the radial residuals from a best-fit circle to a sample of 360 points around the equator of a calibration sphere. Departures from exact 120° symmetry, such as the small shoulders on two of the lobes, arise from the details of the particular probe under study and are not included in the present model. In this work we restrict ourselves to simple straight styli, oriented vertically. Extension to horizontal styli is straightforward.

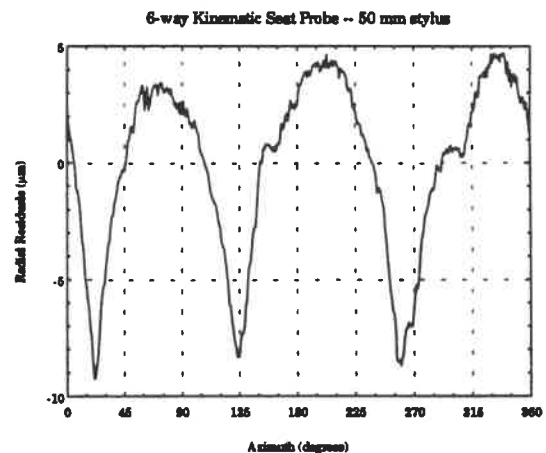


Figure 2. Typical touch trigger probe lobing

The probe design suggests the model shown in Figure 3 which incorporates the essential mechanical features. For an appropriate range of angles θ and φ , a force $\bar{F}(\theta, \varphi)$ will cause the stylus to rotate about an axis AB defined by two of the kinematic seats and the probe will trigger when the tripod leg at C lifts off its contacts. Depending on the direction of $\bar{F}$, two other rotation axes are possible, BC and AC; because of the symmetry we analyze only one lobe. The angle φ_0 determines the phase of the lobing pattern in the CMM coordinate system (xyz); in our analysis, the stylus rotation is constrained to lie in the half-plane defined by $\varphi_0 = \text{constant}$ (i.e., the probe orientation is fixed in the machine coordinate system). In addition to this rotation, the force will also cause the stylus to bend. The total probe tip displacement is the vector sum of the rotational and bending displacements, and the resultant pre-travel is the projection of the total displacement vector onto the direction of motion. The central result of our analysis is the pre-travel function $t(\theta, \varphi)$ given by:

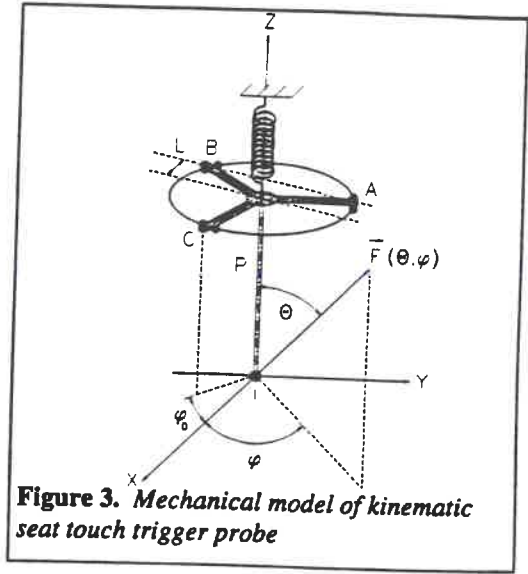


Figure 3. Mechanical model of kinematic seat touch trigger probe

$$t(\theta, \varphi) = \alpha G(\theta, \varphi) + \frac{\gamma \sin^2 \theta}{G(\theta, \varphi)}, \quad (1)$$

where

$$G(\theta, \varphi) = \sin \theta \cos(\varphi - \varphi_0) + \beta \cos \theta. \quad (2)$$

The parameters α , β , and γ are related to the material properties and geometry of the probe construction by:

$$\alpha = P\tau, \quad \beta = \frac{L}{P}, \quad \gamma = \frac{P^2 \sqrt{P^2 + L^2} \cdot F_m}{3EI}. \quad (3)$$

Here, P is the stylus length, L is a small rotation lever arm, τ is the minimum angle through which the tripod must rotate in order to trigger the probe, F_m is the minimum force that will trigger the probe, E is the elastic modulus of the stylus, and I is the area moment of inertia of the stylus transverse cross-section. Equations (1) and (2) are restricted to the azimuthal angular range $\varphi_0 - 60^\circ \leq \varphi \leq \varphi_0 + 60^\circ$ and describe the lobe centered at $\varphi = \varphi_0$. The other two lobes follow by analogy by replacing φ_0 by $\varphi_0 \pm 120^\circ$.

An additional feature of our model follows from the observation that both the pre-travel and its variation are markedly reduced when probing at high angles of incidence (small polar angles θ). For most probe trigger events, displacement of the probe tip is accompanied by sliding on the part surface, a motion that is resisted by the frictional force between the stylus ball and the surface. As the polar angle decreases, the normal force (and hence the frictional force) increases until at some critical angle θ_c sliding is prevented and the probe triggers abruptly. Our analysis of this behavior leads to the expression:

$$\theta_c = \tan^{-1}(\mu_s), \quad (4)$$

where μ_s is the coefficient of static friction between the stylus ball and the part surface. Materials data tables give values for μ_s in the range 0.15 - 0.20 for hard materials on steel, so we expect $8.5^\circ \leq \theta_c \leq 11.3^\circ$; observed values for θ_c are near 10° . When fitting probe data to our model, then, we use Equations (1-2) for $\theta > 10^\circ$ and otherwise we set the pre-travel $t(\theta, \phi) = \kappa$, a constant model parameter to be determined in the fitting procedure.

Test of Lobing Model 6-way Probe - 50 mm Vertical Stylus

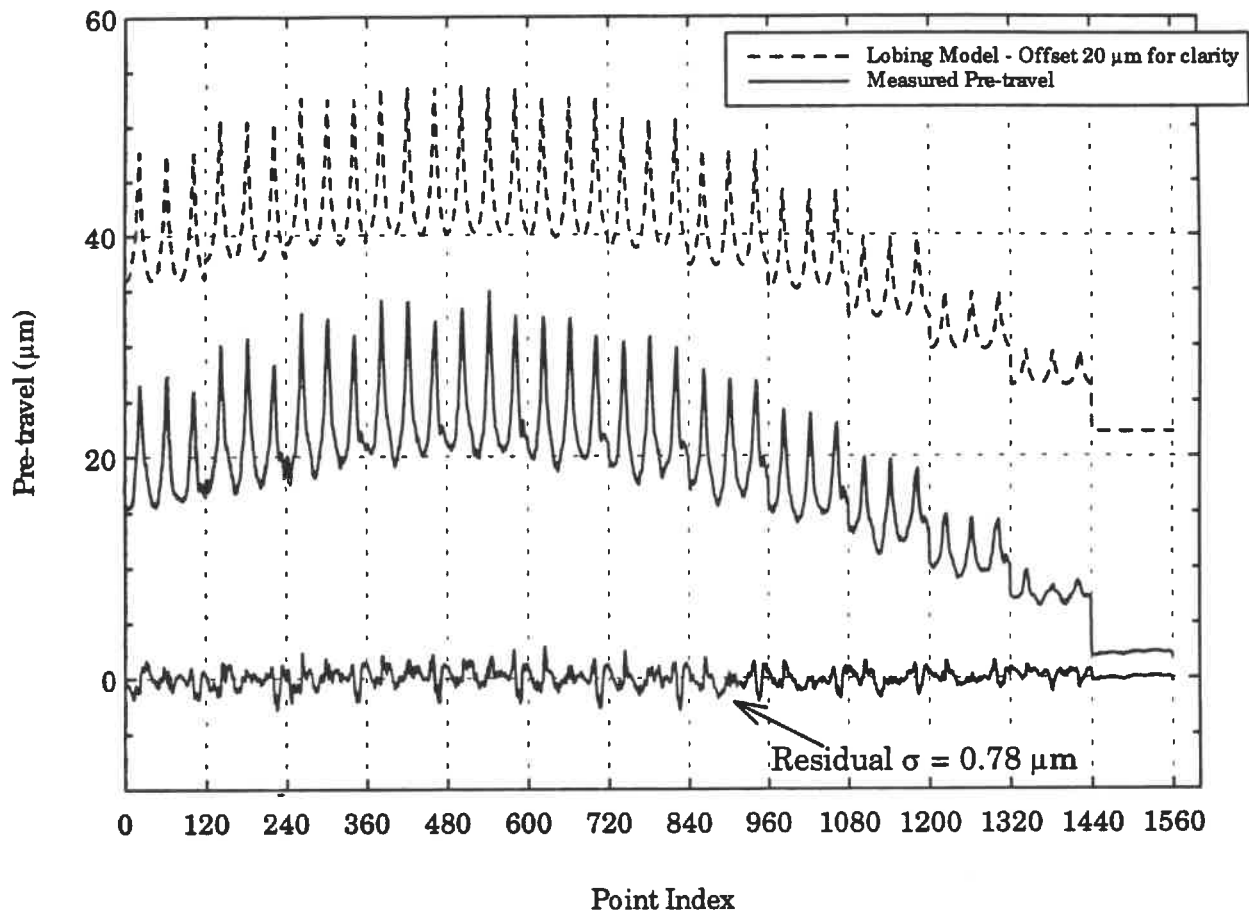


Figure 4. *Least-squares fit of lobing model to measured pre-travel*

Figure (4) shows the results of a non-linear least-squares fit of our pre-travel model to a large sample of data over a high-accuracy calibration sphere. The probe is a commonly-used "6-way" design fitted with a 50 mm stylus extension and a minimum triggering force of 6 grams (0.06N). The sphere was sampled at 120 points around circles of constant latitude in 10° increments from 130° to 10° , with one sample taken at the pole, $\theta = 0^\circ$, for a total of 1561 points per data run. The lobing behavior is clearly evident in this data, and one can clearly see the flat "friction cap"

over the last 121 points. The plotted data are the average of 20 runs, in order to suppress non-repeatable errors and isolate the systematic errors that we are trying to model. The points are plotted against an index which is simply a point number or counter.

The center trace in the figure shows the experimental data, the dashed upper trace (offset by 20 μm for clarity) shows the fitted pre-travel model, and the lower trace displays the difference (fit residuals). The fit yields nine parameters: the center coordinates and radius of the best-fit sphere: (x_c, y_c, z_c, r) , and the five parameters of the probe model: $(\alpha, \beta, \gamma, \varphi_0, \kappa)$. As can be seen, the fit of the model to the data for this probe is very good, with no obvious sign of residual systematic error. The standard deviation of the residuals after correction is 0.78 μm for the averaged data.

For a typical individual run in this experiment, the standard deviation σ_{data} of the uncorrected pre-travel data, about a mean of 17.5 μm , is approximately 7.3 μm , while the residuals after correction are typically distributed about mean zero with $\sigma_{residual} \approx 1.2 \mu\text{m}$. The pre-travel model thus reduces the probe-related uncertainty of measured CMM point coordinates by a factor of about six.

Acknowledgments

We wish to thank D. Ward for assistance with the illustrations. This work was supported in part by the NIST Computational Metrology Program and the Air Force CCG program.

References:

1. McMurtry, D.R., U. S. Patent 4,138,823 "Probe for Use in Measuring Apparatus", Feb. 13, 1979.
2. Butler, C. (1991) "An investigation into the performance of probes on coordinate measuring machines", *Industrial Metrology* 2, pp 59-70.
3. Jarman, T. and Traylor, A. (1990) "Performance Characteristics of Touch Trigger Probes", *Proceedings of Precision Metrology with Coordinate Measurement Systems*, Society of Manufacturing Engineers, Troy, Michigan.
4. Reid, C. (1993) "Probe Technology - Beyond Accuracy", *Proceedings of Applying Imaging and Sensing Technology to CMM Applications*, SME, Nashville, Tennessee.
5. Phillips, S. D. (1995) "Chapter 7: Performance Evaluations", *Coordinate Measuring Machines and Systems*, J. A. Bosch, Ed., Marcel Dekker Inc., New York.
6. Phillips, S. D., Borchardt, B., and Estler, W. T. (1995) "The Estimation of Measurement Uncertainty of Small Circular Features", NIST Interagency Report 5698.

Accuracy Enhancement in Profile Grinding Through Utilization of Post Process Inspection with Feedback

Timothy M. Dalrymple and John C. Ziegert
Machine Tool Research Center
Department of Mechanical Engineering
University of Florida
Gainesville, FL 32611

Introduction

While high-speed CNC profile inspection gauges have been available for over a decade, the potential these gauges present for quality improvement in profile grinding remains poorly utilized. Traditionally, grinding of the profile shape is controlled by precision master cams which generate the relative motion between the workpiece and the grinding wheel. While it is sometimes possible to reduce profile errors through adjustment of process parameters, it is not readily possible to modify the basic profile shape. On traditional profile grinders, such modification can only be accomplished by regrinding the master cams. Recently, however, programmable CNC profile grinders have become widespread. On these machines, the master cams are replaced with programmable profile data. This article describes a control system which exploits the programmable nature of the CNC profile grinders to reduce profile and timing errors. In this approach, part profiles are inspected immediately after the grinding process. Based on the inspection results, the profile grinder's nominal part program data are modified and the next part is ground. In this paper, results of recent experimental work which show an order of magnitude reduction in profile errors with no increase in timing errors are reported.

Description of Approach

The structure of the implemented control system is based on the established industrial convention of specifying the value of the profile at one degree increments. This convention is illustrated in Figure 1. Here, d_i is the distance from the part's axis-of-rotation to the profile surface. In the implemented control system, the 360 individual d dimensions are controlled separately. This controller assumes that the value d_i is independent of d_{i-1} and d_{i+1} . In control system terminology, the d values are said to be non-interacting and the 360 discrete values are treated as separate variables. Also, the timing of the profiles relative to some part feature, such as a keyway, is controlled independently of the profile error.

Profile Geometry: Method of Specification

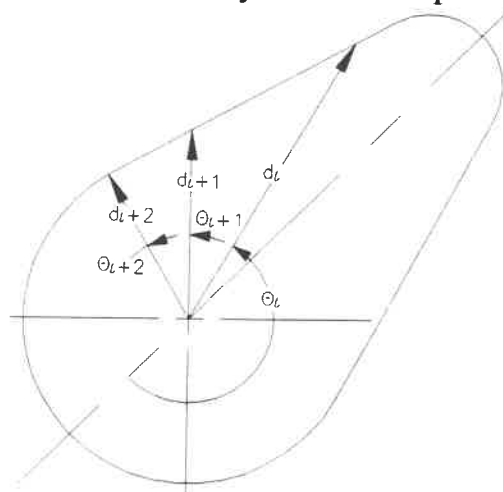


Figure 1

Discussion of Results using a Non-Interacting Process Control Model

To demonstrate the feasibility of this control strategy, a controller based on this simple non-interacting process model was tested. One hundred four-lobe camshafts were ground on a Landis 3L profile grinder and inspected on an ADCOLE 911 cylindrical coordinate measuring machine (CCMM). Figure 2 shows the reductions in the root mean square (rms) of the profile error for six closures of the control loop. Figure 3 clearly shows an order of magnitude reduction in the profile error for lobe 2. Since this control system also regards profile and lobe timing as independent variables, it is important to note that the reduction in profile error was accomplished with no increase in timing error.

Profile Error (RMS)

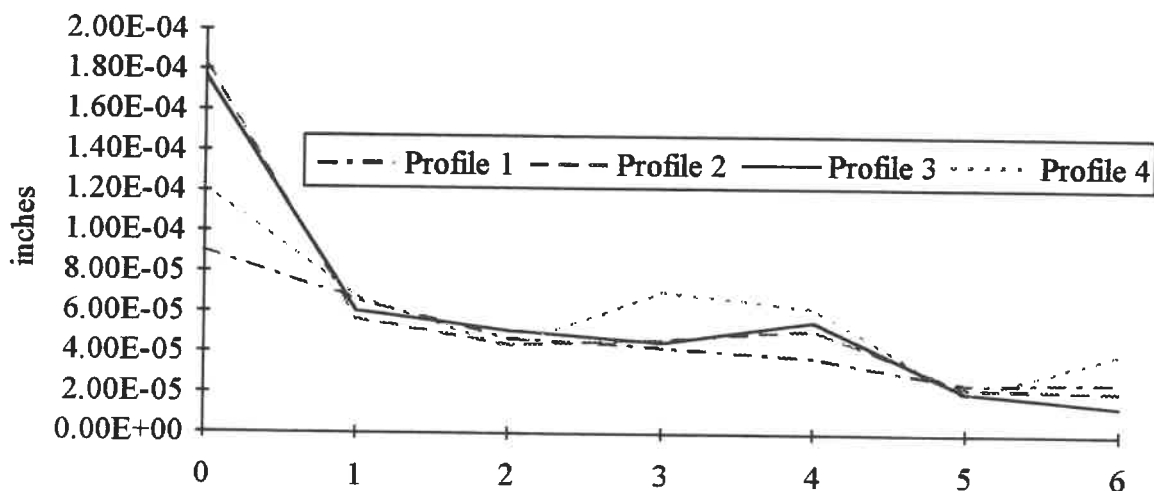


Figure 2

Lift Error for a Typical Lobe

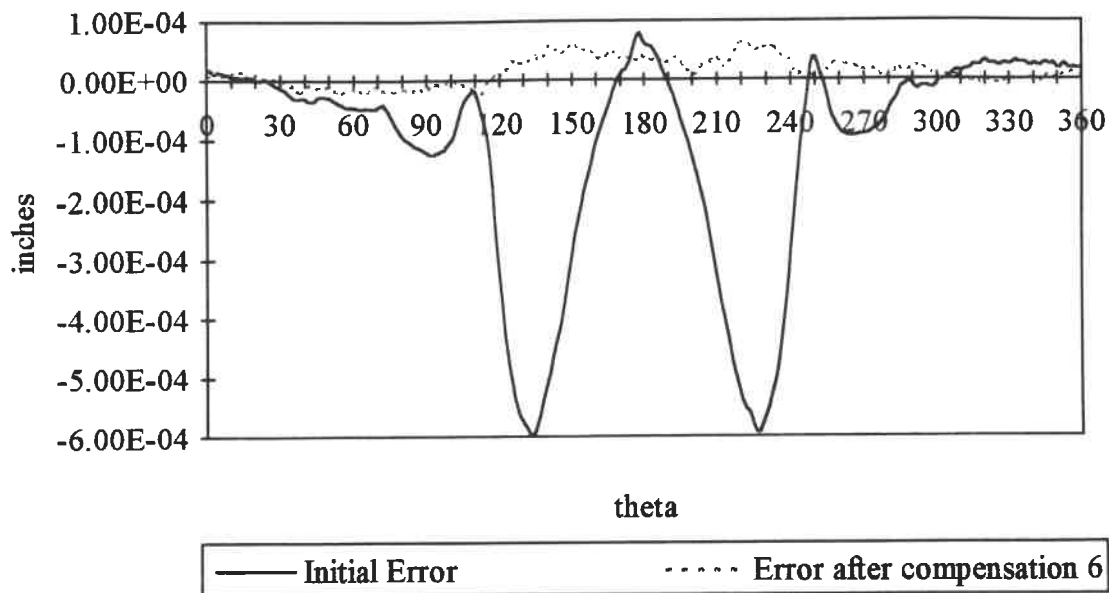


Figure 3

Potential for Improvement in the Rate of Error Reduction

In our implemented control system, we assumed that the d 's do not interact. However, this is not actually the case and using stochastic modeling techniques, the form of the interaction can readily be determined. Using this technique, we fit the measured profile data to a model of the form

$$d_{\text{model}_\theta} = \beta_0 d_\theta + \beta_1 [d_\theta - 2d_{\theta-1} + d_{\theta-2}] \quad (\text{equation 1})$$

where β_0 and β_1 are the model coefficients determined by comparing the commanded part geometry and the resulting ground profile. It's interesting to note that the term associated with coefficient β_1 is simply the geometric acceleration of the profile. As shown in Figure 4, a stochastic process model of this form successfully accounts for 90% of the repeatable component of the profile error.

From this model, a non-interacting control system which compensated for the dependency of the profile values may be proposed. Such a system could compensate for the interaction among the profile values at adjacent points to produce non-interacting control of the profile. Clearly, the ability to control the profile at one point without affecting the profile at adjacent points is limited. However, based on the success of the tested controller and the demonstrated ability to accurately model the process, it should be possible to achieve the same type of reduction while inspecting fewer parts.

Stochastic Process Model

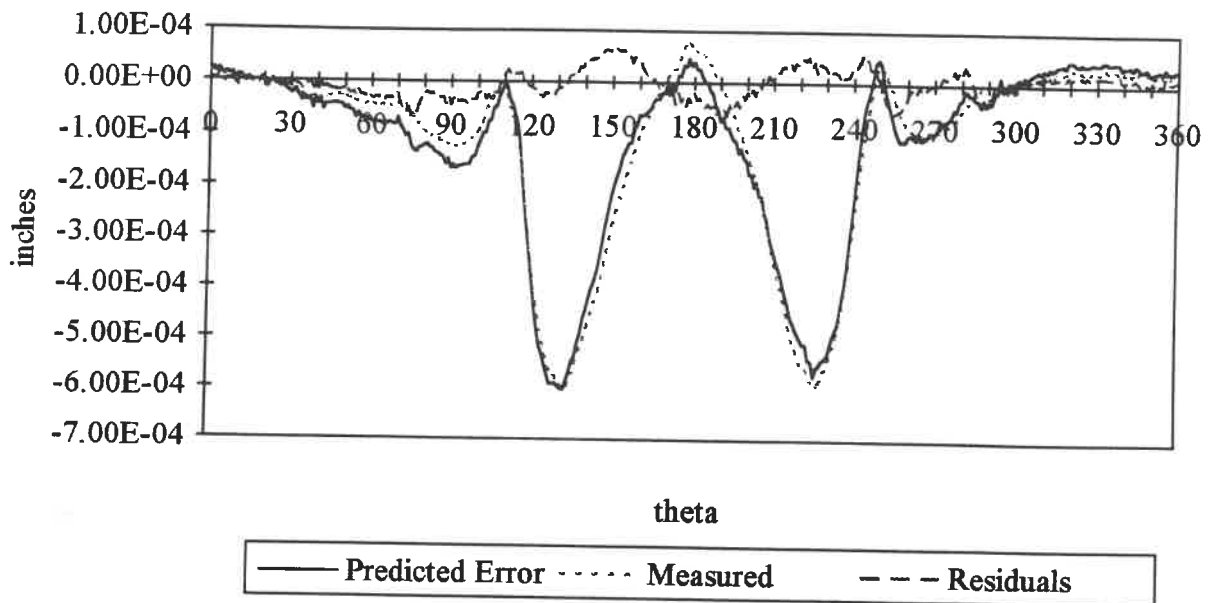


Figure 4

Summary

The results achieved in this work demonstrate the feasibility of a controller for profile grinding based on the feedback of post-process inspection results. Also, the ability to readily model the process error suggests that it may be possible to implement a more effective control strategy which exploits this process model to calculate the required compensation. The implementation of this non-interacting control system is the subject of ongoing research.

Key Words: inspection, feedback, control, CNC, cam, grinding, cylindrical coordinate measuring machine (CCMM), error correction.

References:

- 1: Dalrymple, T. M., "Improved Process Control in Camshaft Grinding through Utilization of Post Process Inspection with Feedback," Master's Thesis, The University of Florida, Gainesville, (1993)
- 2: Bollinger, G. B. and Duffie, N. A., Computer Control of Machines and Processed, Reading, MA: Addison-Wesley, 1988.

Active Thermal Deformation Compensation Based on Internal Monitoring Using a Neural Network with the Genetic Algorithm Method

**Mamoru MITSUISHI, Tsutomu OKUMURA, Naohiko SUGITA,
Takaaki NAGAO and Yotaro HATAMURA**

Department of Engineering Synthesis, Faculty of Engineering
The University of Tokyo
Hongo 7-3-1, Bunkyo-ku, Tokyo 113, Japan

1. Introduction

The main causes of machine tool deformation are heat and force. This paper discusses a method for actively compensating thermal deformation. In the system, position and inclination errors are detected by newly-developed deformation sensors and are actively compensated by actuating the structure of a machining center. The compensation is achieved using 5 degree-of-freedom thermal actuators installed in the lower part of the column. Application of a genetic algorithm integrated with a neural network is also discussed.

2. Related work

The importance of thermal deformation compensation is well known [1, 2, 3], in particular, to realize high accuracy and high reliability. Much research has been performed to overcome this problem. The primary sources, amounts and causes of heat generation have been investigated [4]. High rigidity, heating methods, thermal deformation compensation type structures [5], low thermal conductance materials such as concrete and ceramics, low thermal expansion materials such as low thermal expansion cast iron and heat pipes have all been tried. Thermal control of the factory floor has also been tried, but may be expensive.

Furthermore, there have also been some attempts to compensate for the deformation. Other investigators have used external measurements to determine thermal deformation, for example, laser methods [6]. Attempts have also been made to measure deformation at representative points and predict the induced errors by using a physical model [5, 6, 7, 8]. Some other attempts control the thermal deformation by measuring the oil temperature. Concerning the compensation methods, thermal deformation is used to compensate in some attempts. Other compensation methods include the use of piezoelectric actuators [7] and modification of the software of the controller [6, 8]. There has also been

a study of the use of a neural network to predict the position of the tool. Thermocouples provide an easy method for measuring the temperature. However, this method does not measure deformation directly and, therefore, the output signal may include a large error.

In contrast with the attempts discussed above, this paper reports experiments in which the machine tool deformation is monitored by deformation sensors internal to the machine tool and that the thermal deformation is actively compensated by thermal actuators. Furthermore, a neural network/genetic algorithm was applied for compensation control.

3. Structure of a machining center

To measure the deformation of a vertical type machining center, 4 deformation sensors have been attached to the head, 4 to the upper part of the column and 8 to the lower part of the column, as shown in Fig.1. To actively compensate for the tool position and attitude error, sixteen thermal actuators (matched sets of electrical heaters and cooling jackets) have been attached to the lower part of the column. These thermal actuators do not reduce the rigidity of the system and can easily produce small inclinations because they do not require any bearing elements to produce the motion. By actuating the thermal deformation elements, translational motion in X , Y and Z directions and rotational motion around X - and Y -axis can all be realized.

Each deformation sensor consists of a strain gauge type detecting block and a connecting rod as shown in Fig.2. The deformation between the upper and lower attachment points is concentrated in the deformation of three thin beams located in the center of the detector. Thus, small changes in the long distance between the two measuring points can be detected with high precision and stability. The deformation sensor is made from super-invar metal, with very low

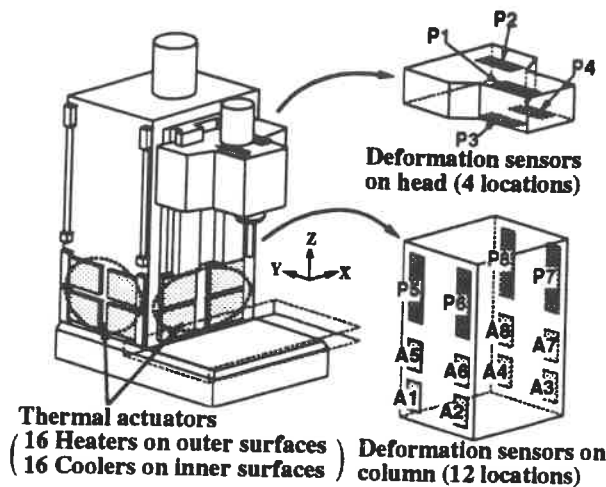


Fig.1 Location of thermal actuators and deformation sensors

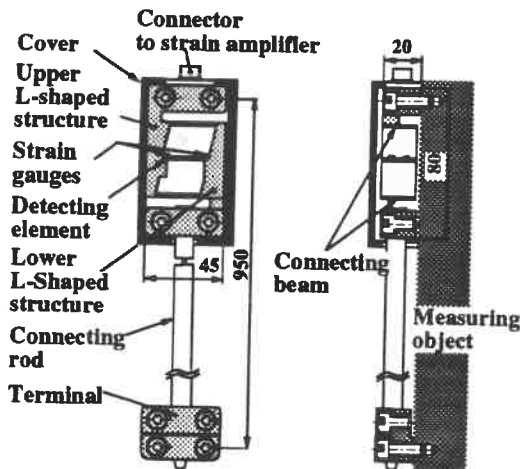


Fig.2 Structure of the deformation sensor

thermal expansion, to ensure that only the thermal distortion of the machine itself is measured without influence of the thermal changes in the environment or of the machine structure. The position of the tool is measured by monitoring the motion of a cylindrical test bar (280mm in length and made of super-invar) using five eddy current sensors.

The performance of the thermal actuators is fast enough to actively compensate for the thermal deformation. It takes approximately 10 seconds to move the tip of the tool in $1\mu\text{m}$.

4. Integration of neural networks and genetic algorithms for tool position prediction

4.1 Prediction of tool position

In the system described in this paper, the orientation error will be compensated by the thermal

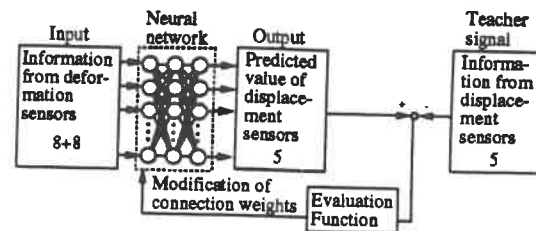


Fig.3 Neural network for the tool displacement prediction

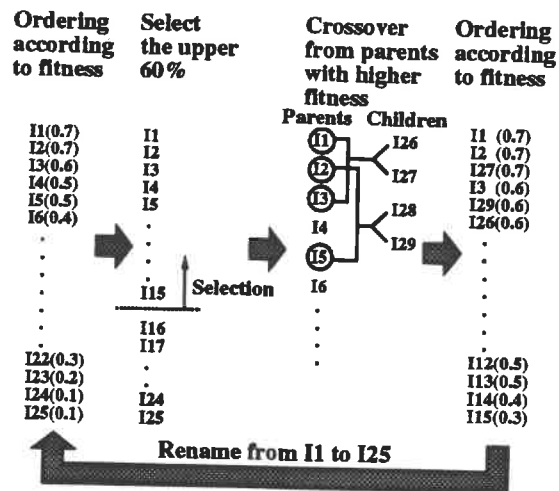


Fig.4 Search using genetic algorithms

actuation method discussed above and translation error will be compensated using the origin shift function of the numerical controller. Therefore, tool position prediction is needed. We have chosen to predict tool position using the output from the deformation sensors to avoid problems in measuring the tool position directly caused by cutting chips and oil. The tool position is predicted by a multi-layer neural network. The inputs to the neural network are the 16 outputs from the deformation sensors and the outputs are position and orientation of the tool as shown in Fig.3. One of the key problems in using neural networks is that the solution may fall into a local minimum, rather than the global minimum. This problem has been addressed by using a genetic algorithm.

4.2 Application of genetic algorithms

Genetic algorithms have been used to reject local minimum solutions (Fig.4). The chromosomes of each gene in a genetic algorithm reflect the connection weights and thresholds of the neural network. The length of a gene is determined as 357 ($16 \times 16 + 16 \times 5 + 16 + 16 + 5$) because the node numbers of input, hidden and output layer are 16, 16 and 5, respectively, considering also the number of thresholds and the number of genes in one

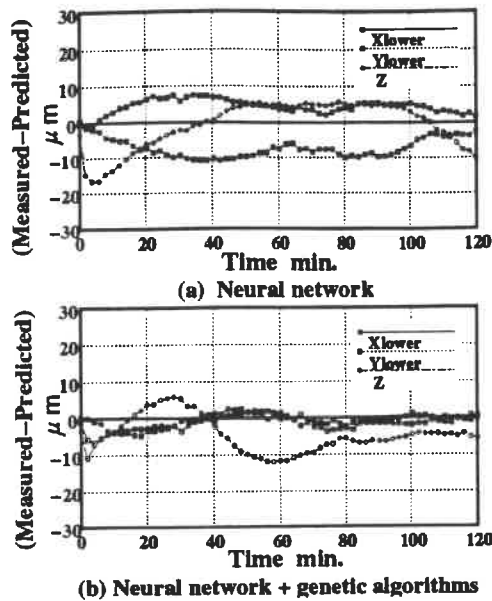


Fig.5 Comparison of NN and NN + GA

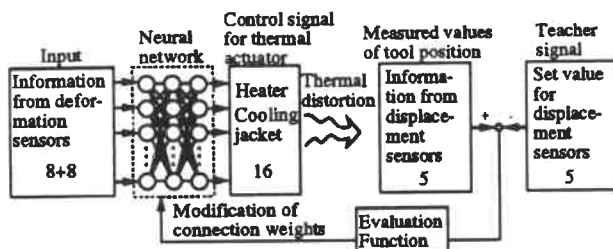


Fig.6 Neural network for learning mode

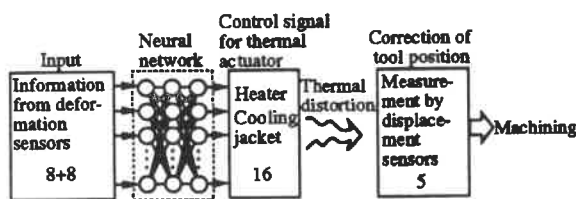


Fig.7 Neural network for control mode

generation is determined to be 25. As an evaluation function which represents the fitting capability of a gene to the environment, an inverse of the summation of difference squares between the predicted values and measured values as teaching signals is used. It is called fitness. The next generation comprises descendants which are generated from the high fitness genes in the current generation. The selection probability is determined as 0.6 in the system. Furthermore, homogeneous crossover method was applied. Mutation probability and the maximum generation were determined as 0.1 and 2,000, respectively. These values concerning genetic algorithms were determined by trial and error.

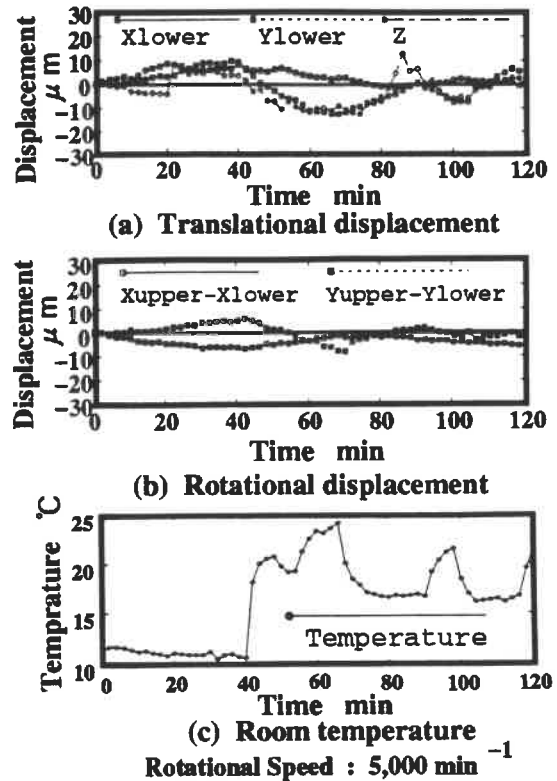


Fig.8 Measured motion of the test bar when thermal distortion is controlled by NN + GA

4.3 Tool position prediction experiments

Fig.5 shows the difference of the tool position prediction capability of the neural network, with and without the incorporation of the genetic algorithm. These results were obtained using identical sensor output data, which was not the data used for the training. The experimental results indicate that the integration the genetic algorithm in the neural network improves the tool position prediction accuracy.

5. Active thermal deformation compensation

To compensate for the tool orientation error caused by thermal deformation, neural networks as shown in Fig.6 and Fig.7 are used. Because it is difficult to measure the tool position and orientation during a cutting operation, the neural network has to have two modes: one is the "learning mode" and the other one is the "control mode" which is used during the actual machining operation. In the control mode, control is executed using the weights and thresholds determined during learning mode operation. The following equation is used as an evaluation function and learning is executed to minimize the

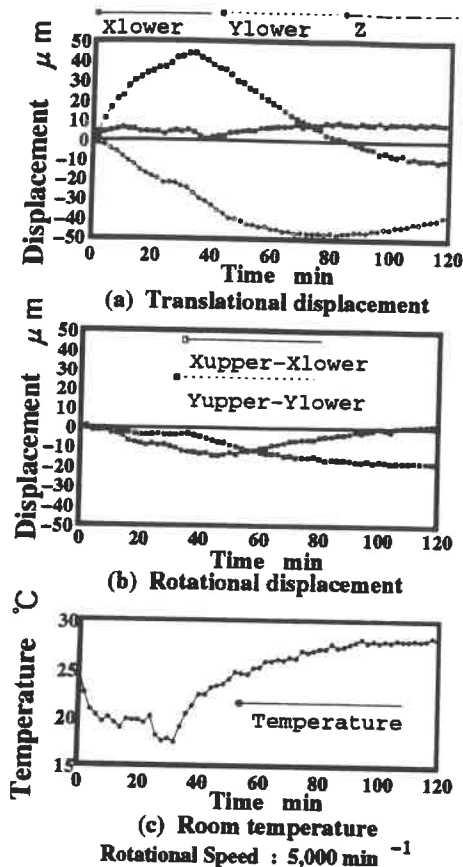


Fig.9 Measured motion of the test bar for uncontrolled machine

evaluation function.

$$E = \frac{1}{2} \sum_c (z_c - \hat{z}_c)^2 \quad (1)$$

where z_c , $\hat{z}_c$ and suffix c represent the current tool position and orientation, the tool position and orientation at the beginning of the experiment, and one of five measured positions of the test bar, respectively. The learning strategy is determined as follows:

$$\Delta w_{kj} = -\alpha \frac{\partial E}{\partial z_c} \frac{\partial z_c}{\partial y_k} \frac{\partial y_k}{\partial u_k} \frac{\partial u_k}{\partial w_{kj}} \quad (2)$$

where k, j, u_k, y_k and w_{kj} represent the output layer node, the hidden layer node, the summation of the outputs from the hidden layer multiplied by the weights and the connection weights between the hidden layer and the output layer, respectively. α is a positive constant. The value of $\frac{\partial z_c}{\partial y_k}$ is determined in advance by considering the relationship between the output of thermal actuators and the change of tool position and orientation. A 3-state extended sigmoid function

is introduced because the output of the network may take on one of three values such as heating, cooling and no operation.

Fig.8 shows the experimental results for tool position and orientation using the methods described in this paper and Fig.9 shows the measured motion of the test bar without control. The rotational speed of the main axis was 5,000 min.⁻¹, the room temperature varied in a range of 14 degrees centigrade, and the position and the attitude of the spindle were maintained within 12 micrometers and 21.4 microradians (6μm/28mm), respectively. The achieved error is less than 30 percent of the level of the uncontrolled machine.

6. Conclusions

A machining center that employs thermal actuators to actively compensate for thermal deformation has been developed. The system measures thermal distortion through an internal monitoring method using newly investigated deformation sensors, and develops compensation strategies using a 3-layer neural network integrated with a genetic algorithm. With a 14 degree centigrade variation in the room temperature, the system maintains the position and orientation of the spindle within 12 micrometers and 21.4 microradians, respectively. The achieved error is less than 30 percent of the level of the uncontrolled machine.

References

- [1] Bryan, J.B., "International Status of Thermal Error Research," *Annals of the CIRP*, 16, 203-215, 1968.
- [2] Tlustý, J., et al., "Testing and evaluating thermal deformations of machine tools," *Proc. 14th MTDR Conf.*, pp.285-297, 1973.
- [3] McKeown, P.A., et al., "Reduction and Compensation of Thermal Errors in Machine Tools," *Annals of the CIRP*, 44, 2, 1995.
- [4] Jedrzejewski, J. and Modrzycki, W., A New Approach to Modelling Thermal Behavior of a Machine Tool under Service Conditions, *Annals of the CIRP*, 41, 1, 455-458, 1992.
- [5] Spur, G., Hoffman, E., Paluncic, Z., Benzinger, K. and Nymoen, H., Thermal Behavior Optimization of Machine Tools, *Annals of the CIRP*, 37, 1, 401-405, 1988.
- [6] Venugopal, R. and Barash, M., Thermal Effects on the Accuracy of Numerically Controlled Machine Tools, *Annals of the CIRP*, 35, 1, 255-258, 1986.
- [7] Moriawaki, T., Thermal Deformation and Its On-Line Compensation of Hydrostatically Supported Precision Spindle, *Annals of the CIRP*, 37, 1, 393-396, 1988.
- [8] Okushima, K., Kakino, Y. and Higashimoto, A., Compensation of Thermal Displacement by Coordinate System Correction, *Annals of the CIRP*, 24, 1, 327-331, 1975.

COMPENSATION OF ERRORS DETECTED BY PROCESS-INTERMITTENT GAUGING

Herbert T. Bandy, David E. Gilsinn
National Institute of Standards and Technology
Gaithersburg, MD

INTRODUCTION

This paper describes a software design for implementing compensation of errors detected by process-intermittent gauging. This is the function of one control loop in the three-loop Quality In Automation (QIA) architecture that is under development at the National Institute of Standards and Technology (NIST) [Donmez, 1992]. Process-intermittent error compensation is useful for improving the accuracy of metal cutting processes on turning or machining centers [Bandy, 1991]. During machining processes, errors due to tool wear, tool deflection, part deflection, etc., are independent of the accuracy of the machine tool itself, and can, therefore, be measured using the machine tool. The approach discussed here is to reduce those process errors.

The term *process-intermittent* refers to inspection occurring during an interval between the semifinish and finish machining processes. Although *in-process* measurement techniques may allow uninterrupted machining [Fan, 1991], process intermittent inspection has advantages. For instance, a simple measurement device such as a touch-trigger probe may be used instead of the more intrusive apparatus required for measurement during active machining. Also, since a complete profile of the error pattern may be obtained in advance of finish machining, computational filters may be applied to the compensating tool path to minimize the effects of measurement irregularities.

The software design facilitates and integrates all pertinent procedures with systematic methods for producing nominal part data, analyzing inspection results, and specifying an adjusted tool path. The main emphasis in this paper is on a system of part feature data generation and management which streamlines the implementation of process-intermittent error compensation. A prototype of the system, tested on a turning center, proved that the employed methods are successful and efficient [Bandy, 1995].

SYSTEM DESCRIPTION

Figure 1 shows the major components of the architecture used. For dimensional error compensation, the part is inspected by on-machine probing after semifinish machining. The system compares the inspection results with the nominal part feature dimensions provided by a CAD system, and computes an adjusted tool path for the finish cut. One option for adjusting the tool path is for the NC controller to execute a part program containing new coordinates calculated for the finish cut. Another option is to use a smart interface to the NC controller to adjust the tool position continuously during machining. For this purpose, the prototype system used the Real-Time Error Corrector (RTEC) [Yee, 1992a], an electronic device developed at NIST to change the position feedback signal before it reaches the NC controller. During machining of the finish cut, feature-specific compensation data are active for each feature in turn. Surface quality may also be inspected after the semifinish cut. Within certain limits, the feed rate and cutting speed will be adjusted to improve surface quality for the finish cut. The functioning of the system is clarified in the next sections.

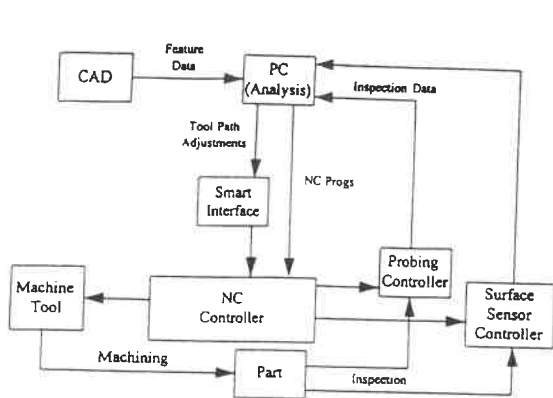


FIGURE 1: System Architecture

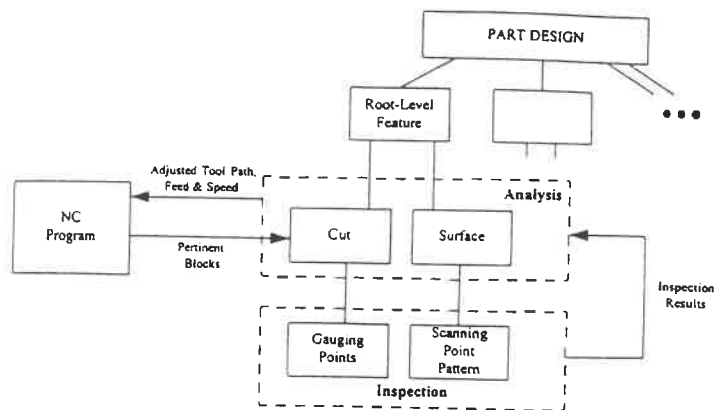


FIGURE 2: The Flow of Feature-Based Data

Data management is an important key to successful error compensation. It is critical to formally record all data associated with design specifications, manufacture, inspection procedures, and inspection results for each part. Especially for post-process analysis, an important phase of the overall QIA strategy which is beyond the scope of this paper, part error data must be traceable to all aspects of the part history. Therefore, the software design for process-intermittent error compensation includes strict means of systematically identifying data (e.g., a probe reading) and relating it to other data (e.g., the corresponding nominal coordinate in a feature). Data labeling nomenclature has been developed to comprehensively serve the data identification needs of the various software programs.

PART FEATURES

The error compensation strategy requires interpreting the part as consisting of separate *features*. Such decomposition of the part is especially useful for establishing correspondences between design information and manufacturing operations [Gupta, 1995]. The rest of this section presents a unique view of part features which not only specifies nominal geometry, but also defines the actual geometry to be a consequence of the machining and measurement processes.

A part feature is a specified region or location on the part design which affects the evaluation of the finished part. Whether areas being machined or inspection points, features are related to others in a hierarchical fashion. Figure 2 illustrates the relationships and indicates how data related to the features are used for error compensation. The basic features which comprise the form of the part are called "root-level" features. Errors are compensated by controlling the machining process for each root-level feature as it is being produced. As explained below, subordinate features are physical aspects of parent features defined in relation to machining or inspection. They determine the final evaluation of the parent feature. To date, a prototype system has been developed for a turning center only. Accordingly, part geometry is defined completely in two dimensions. Therefore, the root-level features are the *arcs* and *lines* which comprise the CAD profile of the part. Several provisions have been made to add more complex curves, but to date, arcs and lines are the only valid root-level features.

A *cut* is a feature type which is defined as the portion of part geometry which is produced by the pass of the tool from its last position to the position specified in a particular block of NC code. As such, rather than being identifiable by its geometry, this type of feature is determined according to the machining

process which produces it. Although a cut is not always a geometric subset of another feature, it is always associated with a feature which is considered to be its parent. The nominal geometry of a cut is identical to the corresponding portion of its parent feature, a root-level feature. This is also true for a *surface* feature, a bounded area on a root-level feature whose roughness is of concern.

An individual *gauging point* is another feature type. The cut on which a point exists is its parent. A cut may have any practical number of gauging points, or none at all. A *scanning point pattern* is a set of many points defined collectively as a single feature. Its parent is the surface on which it is located. The scan mode for the pattern determines how many points will be scanned between designated end points. The mode specifies the time interval if the points are equally spaced in time; or the number of points if they are equally spaced in distance.

Different attributes are associated with different types of features. A root-level feature has as attributes the principal types of errors to which the feature is subject. Identifiers of the initial block and final block of NC code which define a cut are attributes of the cut. The identifiers are integers denoting variables in the blocks of NC code. The cutting parameters are the attributes of a surface to be scanned. The components of the surface normal unit vector at a gauging point are attributes of the point. The scan mode and interval specification are attributes of a scanning point pattern.

PROCEDURES

Process-intermittent error compensation is a multistage process. It begins by segmenting and adapting the original NC part program. The finish cut segment is a template which is generated by replacing tool-path coordinates with variables, and by adding variable tool-offset update commands before the cutting commands. CAD-based methods facilitate the creation of pre-process data such as part feature geometry, nominal coordinates of gauging points, surface normal vectors, and the NC probing program segments.

The part is measured by on-machine gauging after a semifinish cut which uses cutting parameters (speed, feed and depth of cut) similar to those for finish cut. The gauging data are sent to a computer. For each gauging point, an error is computed as a vector from the nominal coordinate to the detected surface.

Least squares curve fitting is used to determine the compensating tool path curve. Coordinates along the nominal tool path for a feature are adjusted—shifted by the magnitude of the calculated error vectors, but in the opposite direction. The compensation curve, derived from fitting the adjusted coordinates to a polynomial, becomes the tool path for the corresponding feature for the finish cut. The adjusted position, orientation, form and dimensions of the feature are thus determined. For cases in which the endpoint of the tool path for a feature also serves as the start point of the tool path for the next feature, the adjusted coordinates for that point are calculated as the intersection of the compensation curves for the two paths. The case in which the two features must have the same slope at that intersection requires that condition as an additional constraint. If the shape of a line, contour or circular arc is found to be deformed with respect to the nominal shape, the interpolation codes in the part program may be replaced by explicit linear point-to-point paths whose endpoints may be modified accordingly. Based on this information, errors may be compensated using either of two options: (1) an error specification file may be used to compensate the errors in real time (the prototype uses the RTEC to accomplish this); or (2) the finish cut NC program may be modified by either changing existing coordinates or replacing individual instructions with a series of linear path instructions.

VALIDATION

A set of experiments was conducted with a tool having a flat spot ground onto it to simulate extreme tool wear. When that tool was used to machine a hemisphere, a ridge appeared as an error. Using the system, the error on the hemisphere was reduced from about 25 μm to within 4 μm [Yee, 1992b]. Several linear "errors" of various magnitudes between 0 and 150 μm were artificially created on another series of parts to test the effectiveness of the error compensation system. The NC part programs for both the semifinish and finish versions of the parts contained slight tapers on face and length features to simulate errors. The nominal part data used by the compensation software did not contain the tapers. Therefore, after probing the semifinish part, the software perceived the tapers to be errors. Both options for implementing compensation were found to be equally effective. Overall, gauging the finished parts on the machine showed residual errors within three times the machine tool resolution (1 μm) plus the standard uncertainty of the probe (1 μm). For small (5 μm) errors, the reduction was sometimes small or not significant.

CONCLUSIONS

The prototype of the described software design facilitated implementation of process-intermittent error compensation on a turning center. The result was an on-line, automated and flexible system. The test results validate the feasibility of the concepts used for compensating errors. The design is also extensible to three-dimensional machining.

REFERENCES

- Bandy, H.T.; and Gilsinn, D.E. *Process-Intermittent Error Compensation for Machine Tool Control*, National Institute of Standards and Technology (U.S.) NISTIR to be published; 1995.
- Bandy, H.T. "Process-Intermittent Error Compensation," in *Progress Report of the Quality in Automation Project for FY90*, M.A. Donmez (ed.); National Institute of Standards and Technology (U.S.) NISTIR 4536; 1991 March.
- Donmez, M.A. *Development of a New Quality Control Strategy for Automated Manufacturing*, Manufacturing International—1992, ASME; 1992.
- Fan K.C.; and Chao, Y.H. *In-Process Dimensional Control of the Workpiece During Turning*, Precision Engineering, 13(1): 27-32; 1991.
- Gupta, S.K.; Regli, W.C.; and Nau, D.S. *Manufacturing Feature Instances: Which Ones to Recognize?*, National Institute of Standards and Technology (U.S.) NISTIR 5655; 1995 July.
- Yee, K.W. *Alternative Designs of a Real-Time Error Corrector for Machine-Tools with "Encoder" Position Feedback*, National Institute of Standards and Technology (U.S.) NISTIR 4832; 1992a April.
- Yee, K.W.; Bandy, H.T.; Boudreaux, J.C.; and Wilkin, N.D. *Automated Compensation of Part Errors Determined by In-Process Gauging*, National Institute of Standards and Technology (U.S.) NISTIR 4854; 1992b June.

Authors Index

Ahbe, T.	76
Ahn, W. J.	84
Aikens, David M.	260
Ananda, Mysore V.	364
Aoyama, Hisayuki	384
Arcona, Chris.	13
Asai, Ryosuke	317
Ascanio, Gabriel	72, 104
Auer, Frank	207
Baldwin, J. M.	388
Bandy, Herbert T.	440
Bartlett, P. N.	368
Beriet, C.	368
Bifano, Thomas G.	191
Blaedel, Kenneth L.	159
Bonnema, G. M.	207
Bonse, Marcus H. W.	35, 354
Borchardt, Bruce R.	428
Boudreau, Brian D.	120
Bowen, D. Keith.	350
Bowman, James L.	100
Brown, C. W.	388
Brown, Norman J.	260
Busch-Vishniac, Ilene	305
Caber, Paul J.	120
Caggiano, Helen	191
Carlson, David	203
Carpinetti, Luiz C. R.	219
Cassou, R. M.	388
Castro, B.	72
Cerino, J. R.	358
Chang, In-Bae	328
Chatwin, S.	72
Chen, Jun	116
Chen, Kuo-Shen	199
Chetwynd, Derek G.	350, 368, 396
Choi, C. K.	68
Choudhury, S. K.	292
Chu, Xiaohua	132
Cunningham, Joseph P.	17
Cuttino, James F.	332
Dai, YuZhong	412
Daito, Michimasa	167
Dalrymple, Timothy M.	432
Damazo, Bradley N.	345*
Dave, Chirayu	280
Davies, Matthew A.	235
Davis, L. A. J.	350
Day, Robert D.	264
de Lange, J.	207
Demarest, Frank	187

Devereux, Owen F.	247
Dixson, R.	+
Dong, Qiwei	231
Dornfeld, David A.	159, 239
Dorval, Rick K.	203
Dow, Thomas A.	13, 243, 332
Du, Junwen	132
Duduch, Jaime G.	219
Dunn, Daniel E.	358
Eberhardt, Keith R.	428
Egert, Charles M.	380
Estler, W. Tyler	428
Evans, Christopher J.	235
Fannar, Heimir	100
Farabaugh, Fred	424
Friedrich, Craig R.	223, 280, 284
Fu, J.	25
Galindo, E.	72
Gao, Robert X.	223
Gao, Yingjie	48
Gardner, J. W.	368
Gargas, Julie A.	203
Ge, Ping	128
Gee, Anthony E.	219, 345*, 416
Genin, Joseph	231
Gilsinn, David E.	440
Grobsky, Kevin D.	187
Gutierrez, Hector M.	301
Hale, Layton	272, 408
Han, Dong-Chul	328
Hannah, Philip R.	264
Hara, Shigekazu	183
Harper, Kari K.	235
Hasche, Klaus	76
Hashimoto, Hiroshi	171
Hatakeyama, Masahiro	376
Hatamura, Yotaro	317, 376, 436
Hatch, Douglas J.	264
Hazelton, Andrew J.	420
He, Tian	305
Henderson, David	215
Henke, R. P.	388
Hercher, Michael	203
High, Martin	268
Hocken, Robert J.	424
Hollis, Ralph L.	372
Hopp, T.	428
Howard, Lowell P.	143
Huang, Peisen S.	112
Hylton, K. W.	380
Ichiki, Katsunori	376

*Extended abstract not available at press time.

*Abstract not available at press time.

Idowu, A.	416
Ikawa, Naoya.	21
Imai, Kenichiro.	171
Inoue, Ryosuke	21
Iwata, Futoshi	384
Jain, Vijay K.	292
Jalkio, Jeffrey A.	364
Jasinevicius, Renato G.	219
Jenkins, Hodge E.	251
Jing, Fangsheng	60
Johnson, Doug E.	187
Jokiel, Jr., Bernhard.	313
Joshi, Ghanashyam A.	25
Jouaneh, Musa	128
Joung, Yoon.	328
Jung, Sung-Cheon	328
Kahl, W. Keith	380
Kamiya, Kazuhide	52
Kanai, Akira	167, 344
Kato, Shigeo	336
Kikkeri, Bharath	284
Kikuya, Yukio.	179
King, T. G.	80
Kitahara, T.	346
Kodera, Sunao	183
Kohno, Tsuguo	151
Kong, Qiang	268
Konicek, Larry	364
Kothiyal, M. P.	400
Krulwich, Debra.	408
Kurfess, Thomas R.	251
Kuriyagawa, Tsunemoto	175, 239
Lau, W. S.	68
Lee, G. L.	96
Lee, H. M.	136
Lee, W.B.	124
Levenson, M.	428
Li, Dongguang.	60
Li, Jing.	88, 92
Li, Ya Yun	247
Li, Yan	48
Li, Yugang	88, 92
Liang, Shao-Xiong.	247
Liu, Shugui	88, 92
Liu, Xian Ping	368
Lu, Boyin	56
Ma, Ping	255, 296
Ma, Shumei.	48
Mandák, Josef.	321
Mansur, David	203
Marlar, Troy A.	17
Marsh, Eric R.	322
Martin, Don	43
Martínez, A.	72
Matsui, Shinsuke.	179

Matsunaga, Kazuo	179
McClain, M.	428
McClure, E. Ray	264
McIntosh, Philip.	31
McWaid, Thomas H.	+
Mehlkopp, K.	211
Melkote, Shreyes N.	288
Miguel, Paulo A. C.	80
Miller, Arthur C.	17
Mitsuishi, Mamoru.	147, 436
Miyashiro, Hiroshi	52
Miyashita, Masakazu.	167, 346
Mizutani, Kenichiro	384
Monterio, Andre.	199
Moon, Kee S.	132
Morantz, P.	344
Morrison, Euan	392
Mou, Jong-I	96, 136
Mulholland, George P.	276
Muthu, Ravi	280
Nagao, Takaaki	147, 436
Nagarajan, T.	155
Nagarajan, Thamyliniane	284
Nakao, Masayuki.	317, 376
Narayanasamy, K.	309
Nayfeh, Samir	322
Nesarikar, V.	292
Nijse, Gert-Jan.	35
Nomura, Takashi	52
Obitsu, Naoki.	227
Oezmeral, Hueseyin	211
Ohira, Humikazu	179
Ohta, Takashi	147
Okabe, Hideyuki	147
Okumura, Tsutomu	436
Ouvreloeil, Tita	31
Padilla, S.	104
Pahk, Heui-Jae	84
Palomino, D.	104
Pandit, S. M.	25
Paterson, Robert L.	17
Pawlus, P.	396
Phillips, Steven D.	428
Piscotty, Mark A.	159
Porto, Arthur J. V.	219
Pu, Xinglin	116
Quaid, A.	372
Raja, Jay	120
Ramana, R. Venkata	155
Rao, Balkrishna C.	223
Ren, Ying.	60
Ro, Paul I.	301
Ruiz, G.	104
Sadashivappa, K.	309
Sánchez, José.	104

Sannareddy, H.	120
Sasaki, Akira	384
Sato, Masakazu	167
Scagnetti, Paul	322
Scattergood, Ronald O.	195
Schneir, Jason	+
Scott, Paul J.	392
Shan, Y. Y.	68
Sharp, Keith W.	195
She, Yimin	280
Shen, Y.	428
Shimada, Shoichi	21
Shipley, James C.	276
Shore, P.	344
Singaperumal, M.	309, 400
Sirohi, R. S.	400
Slocum, Alexander H.	322, 345*
Smith, Stuart T.	199, 368
Song, Jin	60
Spronck, Jo W.	35, 354
Srinivasa, Narayan	404
Srinivasa, Y. G.	155
Srinivasan, Nandakumar	424
Stancil, Brent A.	301
Stone, Jack A.	143
Storz II, Gene E.	243
Sugita, Naohiko	436
Summerhays, K. D.	388
Sun, Yunquan	340
Sutedjo, Iwan H.	100
Sutherland, John W.	132
Suzuki, Hirofumi	183
Syoji, Katsuo	175, 239
Takeda, Hiroshi	183
Tanaka, Katsutoshi	183
Tanaka, Shuji	376
Tang, Jianshe	239
Tashiro, Hatsuzo	52
Taylor, John S.	159, 260
Terwei, T.	211
Thiele, K.	76
Tian, Guiyun	64, 140
To, S.	124

Tran, Hy D.	203
Trumper, David L.	199
Tsai, Vincent W.	+
Tsunaki, Roberto	219
Uchikoshi, Junichi	21
Uda, Yutaka	151
Udupa, Ganesha	400
Uehara, Kunio	227
van Beek, H. Frank	207
Van Peski, Chris	+
Wan, De-An	48
Wang, Min	255, 296
Wang, Wanjun	305
Wang, Xu-Jin	175
Wang, Yiding	132
Weaver, L. F.	159
Weck, Manfred	211
Wilhelm, Robert G.	100, 424
Williams, E. D.	+
Williams, Mark	199
Witzgall, C.	428
Xu, Xiaorong	112
Xu, Wei	231
Yan, Bin	64
Yano, Takashi	317
Yau, Hong-Tzong	108
Yazawa, Takanori	151
Ye, Shenghua	56
Yordy, Don H.	408
Yoshikawa, Kazuo	52
You, Zheng	116
Yung, Kai L.	68
Zang, Yanfen	132
Zarudi, Irena	163
Zhang, Bi	340
Zhang, Guoxiong	132
Zhang, Liangchi	163
Zhang, X.	428
Zhu, Hongxi	56
Zhuang, Hanqi	39
Ziegert, John C.	313, 404, 432
Zimmermann, Robert	268
Zou, Zhihua	88, 92

Appendix



A S P E

The following pages are a reprint of the first ASPE Conference Proceedings printed in 1985.

ABSTRACTS

1985 Precision Engineering Conference

14-15 November
Raleigh, North Carolina



North Carolina State University

Lawrence Livermore National Laboratory



1985 PRECISION ENGINEERING CONFERENCE

November 14-15, 1985
Sheraton-Crabtree Inn
Raleigh, North Carolina

Wednesday, November 13

7:00 pm-
10:00 pm Registration Second Floor Mezzanine

7:00 pm-
10:00 pm Reception. President's II Room

Thursday, November 14

8:00 am Registration Second Floor Mezzanine
8:30 am Technical Session I. Governor's Room

 Moderator - C. Teague, National Bureau of Standards
 * Results of Accuracy Testing and Initial Operating Experience of
 the LODTM - D. Atkinson, Lawrence Livermore National Lab. and
 T. Esler, National Bureau of Standards
 * Design and Applications of Ultraprecision Diamond Turning Machine
 - M. Katsuki, TOSHIBA Machine K.K.

10:00 am Break
 * Interferometry With Computer - Generated Holograms
 for Strong Aspheric Surface Testing - T. Yokokura, Tokyo
 Optical Co. Ltd.
 * Damage Effects Indentified by Scatter Evaluation of Supersmooth
 Surfaces - K. Stowell, Wright Patterson AFWAL

12:15 pm Lunch President's II Room
1:30 pm Technical Session I (continued) Governor's Room
 * Polishing Research - N. Brown, Lawrence Livermore National Lab.
 * Application of P-MAC Polishing to Si Wafers - T. Kasai, Saitama
 University

3:00 pm Break
3:30 pm Round Table Discussion, "Organization of an American Precision
 Engineering Society" Governor's Room
 Moderator - R. McClure, Lawrence Livermore National Lab.

4:30 pm N. C. State Precision Engineering Laboratory Tour
7:00 pm Dinner. President's II Room

1985 PRECISION ENGINEERING CONFERENCE

November 14-15, 1985
Sheraton-Crabtree Inn
Raleigh, North Carolina

Friday, November 15

8:30 am Technical Session II. Governor's Room

Moderator - D. Thompson, Lawrence Livermore National Lab.

* Micro-grooving for Plastic Grating Lens with CNC Diamond Lathe -
K. Ueda, TOSHIBA Co.

* An Analysis of Shear Band Behavior in Diamond Turning -
J. Strenkowski, North Carolina State University

10:00 am Break

* Diamond Grinding of Optical Surfaces on Brittle Materials -
L. Chaloux, Pneumo Precision Inc.

* Ultraprecision Grinding Technology for Brittle Materials -
J. Yoshioka, F. Hashimoto, A. Kanai, Tokyo Metro. University,
M. Miyashita, Consultant, T. Abo, Citizen Watch K.K. and M.
Daito, Nissin Machine Works K.K.

12:15 pm Lunch. President's II Room

1:30 pm Technical Session II (continued). Governor's Room

* Design and Applications of Ultrahigh Speed Internal Grinding Head
---- 200,000 r.p.m. - M. Ota, SEIKO Seiki K.K.

* Application of Miniature Bearings to Mechatronics - K. Kakuta,
Nippon Seiko

* Correction of Spindle Accuracy - T. Bifano and T. Dow, North
Carolina State University

3:00 pm Break

3:30 pm Round Table Discussion, "The Education of a Precision Engineer"
. Governor's Room

Moderator - R. Hocken, National Bureau of Standards

RESULTS OF ACCURACY TESTING AND INITIAL OPERATING
EXPERIENCE OF THE LARGE OPTICS DIAMOND TURNING
MACHINE AT LAWRENCE LIVERMORE NATIONAL LABORATORY*

By

D. P. Atkinson
Lawrence Livermore National Laboratory
P.O. Box 808, L-792, Livermore, CA 94550

And

W. T. Estler
National Bureau of Standards
Center for Manufacturing Engineering
METROLOGY A-107, Gaithersburg, MD 20899

The Large Optics Diamond Turning Machine (LODTM) was constructed to diamond turn radially aspheric reflective metal optics up to 64 inches in diameter weighing up to 3000 pounds. It was designed and built at Lawrence Livermore National (LLNL), under sponsorship of the Defense Advanced Research Projects Agency (DARPA). The LODTM incorporates a multiple-path laser feed-back system, capacitance gauges, a 32-bit computer and capstan drives to provide two axes of tool motion in a 32-inch radius by 20-inch length. The design goal for tool positioning accuracy within this work volume is 1.1 microinches rms, with a surface finish goal of 42 angstroms rms.

LODTM is the result of the need for optics that cannot be produce by conventional optical fabrication techniques. Likewise, the figure accuracy of optics of this type cannot be verified by existing equipment. This led to the conclusion that the LODTM must also be used for in-situ metrology. To enable a machine tool to measure its own product, it must be proven to be free of systematic errors that would be repeated during both machining and measuring, effectively hiding the errors.

To accomplish this, DARPA sponsored a joint project by the National Bureau of Standards (NBS) and LLNL to validate the LODTM as a measuring machine. These validation tests were conducted during January-February and November-December of 1984. This paper presents the results of these tests, and shows that the LODTM satisfies the requirements of a measuring machine for the optics to be produced.

During 1984 and early 1985, test samples were diamond turned on the LODTM to determine that surface finishes that could be expected, and to provide samples for evaluating subsequent cleaning and optical coating operations. The results of surface analysis by optical and mechanical techniques is reported, as are some interesting results of measurements of the scattering from the surfaces.

*Work performed under the auspices of the U.S. Department of Energy by the

DESIGN AND APPLICATIONS OF ULTRAHIGH SPEED INTERNAL GRINDING

HEAD-----200,000 r.p.m.

By

Sadao Moritomo

And

Masato Ota
SEIKO Seiki Co., Ltd.
Japan

1. Introduction

Recently in Japan, the demand for precise miniature ball bearings has become very strong. The demand emphasizes the need for smaller size, higher precision and lower cost. The applications are mainly consumer products such as video tape recorders, audio systems and floppy disk drives.

The internal grinding process which is the most important process in the manufacture of miniature ball bearings, determines quality and the manufacturing cost. Because of the recent trends toward miniaturization, bearing manufacturers are faced with a cost reduction barrier, caused by the small workpiece size, insufficient cutting speeds, and insufficiently rigid tools.

The ultrahigh speed internal grinding wheel head recently developed, which rotates at 200,000 rpm, is one of the solution for these problems.

2. Necessity of ultra highspeed rotation

A grinding wheel peripheral speed must be maintained at least 30 m/s (6,000 SFPM) for efficient grinding and speeds of more than 60 m/s are very common in external grinding process. However, it is very difficult to achieve the required peripheral speed, with small diameter grinding wheels. For example, a 1 mm diameter wheel must rotate at 600,000 rpm to achieve a peripheral speed of 30 m/s.

It is generally understood that insufficient wheel peripheral speed increases the normal component of the grinding force, requiring higher grinding pressure to obtain the same metal removal rate as the lower speed. This increased normal force bends the wheel arbor further, and lowers the geometric accuracy. If the infeed speed is decreased, but the grinding force is decreased, but the abrasives slip more easily on the workpiece surface under low wheel speeds and with less rigid slim wheel arbors. The interference angle of an abrasive grain against the workpiece surface of a cylindrical internal grinder is usually smaller than in other grinding process, such as external grinding and surface grinding. Consequently, in small scale internal grinding, the most important factor is to keep the wheel peripheral speed high enough to grind accurately and with high productivity.

3. Characteristics required for internal grinding wheel heads

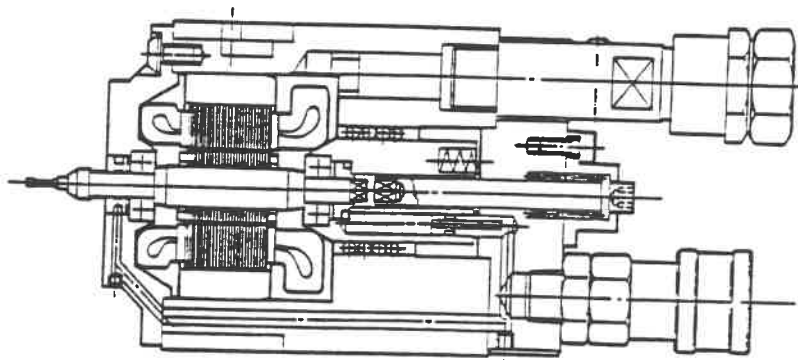
To grind a small size bore precisely, an ultra high rotational speed is an essential characteristic of the wheel head. In addition to this rotational speed, the following characteristics are required for the grinding head.

- a) Ultrahigh rotational speed
- b) Proper output power from the shaft
- c) High static and dynamic stiffness
- d) High rotational accuracy
- e) Smooth and quiet rotation
- f) High wheel mounting accuracy and mounting stiffness
- g) Minimum temperature rise
- h) Long service life

None of these characteristics can be disregarded at the design and manufacturing stages.

4. Design of the 200,000 rpm wheel head

This wheel head we developed to improve the productivity of the internal grinding process, especially for 3 - 4 mm inner race bore miniature ball bearings. A wheel head for this speed range usually has a built-in motor. Only this type of construction satisfies the demands listed above.



The cross-section of the 200,000 rpm wheel head

5. Characteristics of grinding machine

Dynamic stiffness. There should be no resonance up to 3.3 KHz (wheel head driving frequency). If there is resonance, it must be well damped. One of the solutions for this problem is to use a welded steel base (light weight, high stiffness and effective reinforcing ribs). Another solution is a hydrostatic table guide way.

Vibration. All vibration sources except the wheel head must be insulated from the main structure of machine (hydraulic units are separate, work head motor and wheel table oscillation motor are mounted on isolator rubber).

Work fixture. The work piece holding and driving method is very critical. Usually miniature bearing races are supported by a reference shoe and an axial plate, and are driven by two rotating disks. Axial clamping is done by a floating rotor which is suspended by air bearings and is depressed by pneumatic pressure towards the workpiece. By this method, small workpieces can be rotated at over 10,000 rpm.

6. Grinding performance

One example of grinding machine performance involves a miniature bearing with a 2 mm bore. The wheel peripheral speed is still insufficient, but the productivity is 7.5 sec. per piece. In this case the workpiece rotates stably at 10,000 rpm, so a bore out-of-roundness of not more than 0.2 μ m or less can be guaranteed.

7. Conclusion

Each year brings increased demand for cost reduction and improvement in accuracy. Higher-speed, higher-power, vibration free wheel heads are expected of the industry, but today's technology limits the shaft diameter of a 200,000 rpm wheel head to a maximum of 6 mm. This slim shaft in turn limits the stiffness of the wheel head, and the shaft bending frequency. The technology for solving these problem involve high speed bearings as well as higher power and lighter weight driving motors.

One of the most promising bearing technology developments for this purpose is a magnetic bearing using a controlled electromagnet. This magnetic bearing technology allows the shaft diameter to be doubled and permits the mounting of a powerful heavy motor, while decreasing friction losses in the bearing.

INTERFEROMETRY WITH COMPUTER - GENERATED HOLOGRAMS FOR STRONG ASPHERIC SURFACE TESTING

By

T. Yokokura
Chief Engineer
Tokyo Optical Co. Ltd.
Tokyo, Japan

Abstract

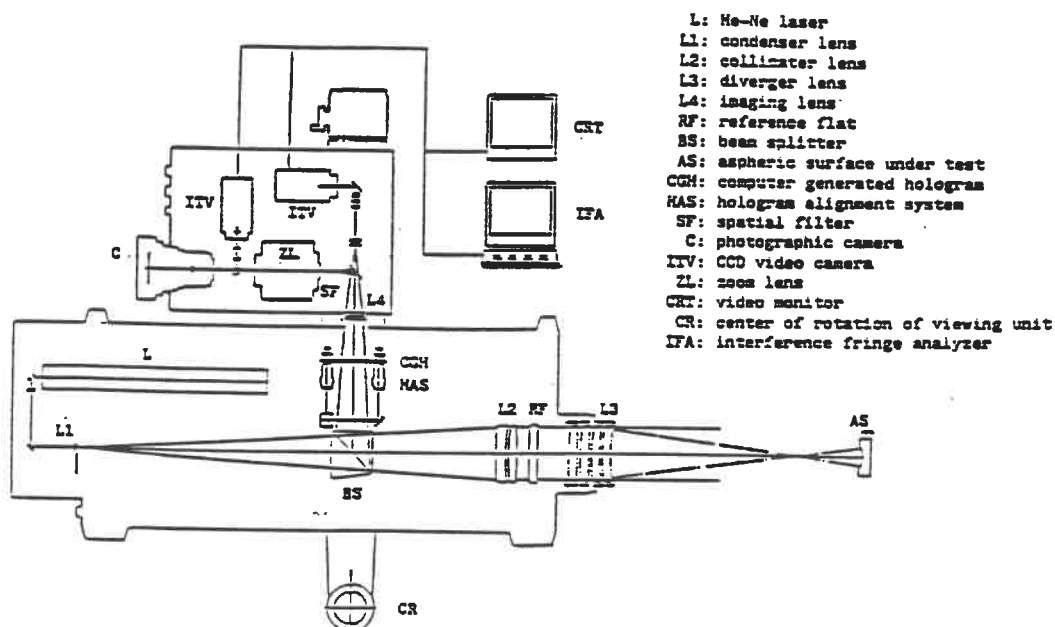
In recent years, aspheric optical elements have gradually come into use, for aiming optical systems lighter and more accurate. In order to bring aspheric optical elements into more common use, however, it is necessary to establish mass production technology and also develop the testing technology for surface profiles accurately and easily.

This interferometer is developed to test aspheric surface, and can be used practically. The feature is as follows.

- 1 A non-contact testing
- 2 Accurate computer-generated holograms which have been realized by electron beam writing are used.
- 3 A unique hologram alignment method is used, enabling accurate alignment to be performed easily and in a short period.
- 4 The fabrication error from the ideal aspheric surface can be observed immediately by means of interferograms.
- 5 Accurate testing of strong aspheric surface is possible.

Maximum aspheric departure is about 1000 waves and testing accuracy is under $0.1\mu\text{m}$.

- 6 An on-axis or off-axis hologram can be selected, hence by changing the F number of the diverger lens appropriately. The interferometer can be used for testing a wide range of curvature and aspheric departure.
- 7 A Fizeau type interferometer has been adopted, enabling both plane and spherical surfaces to be stably and accurately tested.



DAMAGE EFFECTS IDENTIFIED BY SCATTER EVALUATION
OF SUPERSMOOTH SURFACES

By

W. Kent Stowell
AFWAL/AADO-2
Wright-Patterson AFB, Ohio 45433

The surface quality of optics used in an extremely sensitive laser instrument, such as a Ring Laser Gyro (RLG), is critical. This has led to the development of a Variable Angle Scatterometer at the Air Force Wright Aeronautical Laboratories at Wright-Patterson Air Force Base. This device can detect low level light scatter from the high quality optics used in RLG's without the need to first overcoat with metals. With this instrument, damage effects that occur during the typical processing and handling of optics can be identified. Such damage is the cause of the wide variation in scatter measurements depending on when in the process, one takes data. These measurements indicate that other techniques such as Total Integrated Scatter (TIS) may be inadequate due to lack of sensitivity for determining the quality of extremely low scatter surfaces.

The general term for optical surfaces which are better than the lowest level of the scratch-dig standards has become "Supersmooth," and this term is seen in technical literature as well as in advertising. A performance number, such as Bidirectional Radiation Distribution Function (BRDF), which can be measured from the uncoated optical surface by equipment such as the Variable Angle Scatterometer is proposed as a method of generating better optical surface specifications. Data show that surfaces with average BRDF values near 10 parts per billion per steradian (0.010 PPM/Sr) are now possible and measurable.

Recently, the use of BRDF measurements has expanded beyond optical surfaces to semiconductor and even metal surfaces. This method is proving to be of value in many areas of surface and sub-surface analysis. It is suggested that BRDF measurements could provide an effective method of evaluating precision machined surfaces.

POLISHING RESEARCH *

By

Norman J. Brown
Lawrence Livermore National Laboratory
Livermore, California 94550

This paper will attempt to describe the microprocesses involved in the abrasive type of optical polishing process and to correlate it with work currently in progress in Europe, Japan, and the U.S. The emphasis, from the view of an optical fabricator, will be on those processes involving the surface region to a depth of a few hundred angstroms at most. The paper will concentrate on understanding the processes rather than in describing production techniques. It will briefly describe research techniques and modeling.

Surfaces are a three dimensional region that may differ considerably from bulk material in chemistry, structure, and thermodynamic state. These differences may extend from a depth of a few tens of atomic layers to some thousands depending on the material and the process involved. Physically there is a transition from the kinetics of the interior to those of the surface discontinuity, a transition described as an entropy gradient. Chemically, at the surface, bonding no longer extends outward. In addition there are often chemical differences due to diffusion that can extend to considerable depths. There are possibilities of reversible chemical reactions between the base material and diffusing species triggered by changes in the state of strain of the material. During these reactions, structural reorderings can take place, manifested by the appearance of ductility in an otherwise brittle material.

Brown (1981) proposed an abrasive model for this surface region for metals consisting of Hertzian elastic penetration by abrasive particles followed by shear. Brown and Cook (1984) proposed an extension of this to glasses involving a chemical rather than mechanical tooth necessary for shear. This extended model explains why polishing compounds for glasses are limited to those certain ions that enter either a tetragonal or charge compensated tetragonal coordination similar to silicon. Mori, et al. (1985) proposed a similar Hertzian penetration and shear as a result of their elastic emission machining process studies and observed chemical effects similar to Brown and Cook. In aqueous solutions, hydrolyzed silica debris repolymerizes and reabsorbs onto the surface to provide a "bartop" varnish. Neuroth (1985) has observed that this redeposition minus soluble network modifiers would explain the compositional independence he observed in laser surface damage for silicate glasses (1982). Cook (1985) has observed that the effect of stress on the diffusion and solution of water in glass shown by Nogami and Tomozawa (1983) could contribute to a glass surface ductility as a result of the moving stress field in the vicinity of a traveling abrasive particle.

*Work performed under the auspices of the U.S. Department of Energy by the Lawrence Livermore National Laboratory under Contract W-7405-Eng-48.

Application of P-MAC POLISHING TO Si WAFERS

By

Toshio Kasai
Saitama University
Simo-okubo, Urawa, 338, Japan

P-MAC (Progressive Mechanical and Chemical) polishing is newly proposed polishing method for preparing mirror-like, damage-free, optically flat surfaces for high-performance compound semiconductor wafers. This polishing method is characterized by a change in the material removal process from mechanical action to chemical action, as polishing advances. The ideal method for carrying out such polishing must reduce the gap distance between the work and polishing tool. As an example, three particular features can be adopted to change the gap: 1) polishing apparatus or jig, 2) slurry supplying conditions, and 3) work/dummy material conditions.

In P-MAC polishing of GaAs wafers, the principle for generating increased stock removal differences between the work and the glass or sapphire dummy materials has been applied to satisfactorily meet the requirements mentioned above. The roughness and flatness of a finished wafer surface has thus far only reached $0.02\text{ }\mu\text{m}$ in height and $10\text{ }\mu\text{m}$ for a 2-inch diameter respectively. This of course indicates that technically achieving a sufficient degree of polishing is an ongoing requirement for high-quality surfaces. The solution to more effective polishing seems to lie with choosing smoother skinned polishers and better suited polishing solutions.

For the normally prepared Si wafer, an adequate mechano-chemical polishing condition has been produced using colloidal silica-suspended alkaline solution in conjunction with a polyurethane polisher. After the polishing process, however, chemical etching and cleaning are essential for attaining completely satisfactory wafers. By exchanging the conventional polishing method and P-MAC polishing without abrasives, a simplified and economical fabrication process could be expected. In P-MAC polishing of Si wafers, several kinds of active solutions containing fluorine have been tried in conjunction with a plastic dummy material. The accuracy and quality of polished Si wafers are considerably inferior to surfaces which have been mechano-chemical polished. Under these conditions, choosing a polishing solution and a polisher for effecting smoother polished surfaces has been important.

MICRO-GROOVING FOR PLASTIC GRATING LENS WITH CNC DIAMOND LATHE

By

Katsunobu Ueda
Toshiba Machine K.K.
Japan

High quality micro-lenses have been one of the most important elements in laser pick up heads for optical disk systems. A development plan of high quality molded grating lenses was made. The grating lens (shown in Figure 1) is attractive because it has several advantages: (1) Very thin lens, (2) Light weight, and (3) Producibility in large numbers.

The molding die for plastic grating lenses has been made by electroforming. The master disk for electroforming has been machined by diamond turning. The master disk has 220 saw-shaped grooves in 2.6 mm radius. The groove depth is 1.6 μm and its minimum width is 6 μm . The micro-grooves were machined on a lathe numerically controlled by a computer with single crystal diamond tool. Machined master disk material is OFHC-cooper.

Figure 2 is scanning electron microscope (SEM) photographs of the fabricated master disk cross section: (a) near the edge of the disk and (b) the central region of the disk. It was found that 6 μm width and 1.6 μm depth saw-shaped micro-grooves was achieved.

The molded plastic grating lenses have a numerical aperture of 0.13 and focal length of 22 mm. It had the advantage of high efficiency and practicability for fabrication.

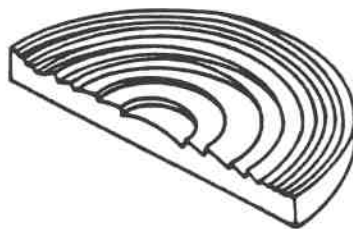
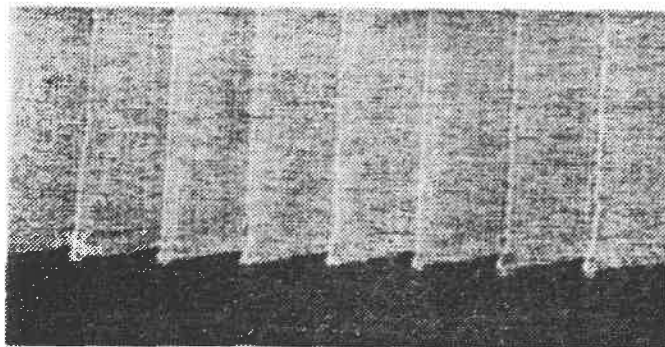
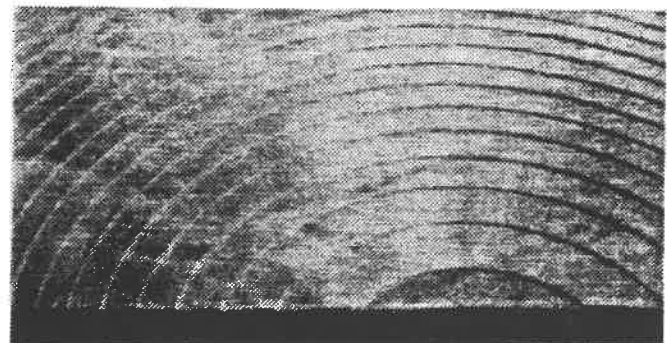


Fig. 1. Grating lens diagram



(a)

10 μm



(b)

100 μm

Fig. 2. SEM photographs of master disk cross section: (a) Near the edge, minimum width 6 μm , (b) Central disk region

AN ANALYSIS OF SHEAR BAND BEHAVIOR IN DIAMOND TURNING

By

John S. Strenkowski
Precision Engineering Laboratory
North Carolina State University
Raleigh, NC 27695

Shear banding is a phenomenon that consists of intense localization of shear along a narrow region in the deforming material. In metal cutting, the presence of a shear band ahead of the tool cutting edge results in the formation of serrated or discontinuous chip. This paper studies the role that shear banding plays in the formation of both discontinuous and continuous chips. Shear banding behavior was investigated using a finite element model of the cutting process. It was found that shear banding occurs if the material exhibits strain--softening behavior. Study of the stress distribution in the region ahead of the shear band revealed that the maximum shear stress reached the instability limit of the material along a line just ahead of the shear zone. This curve also had a similar shape to the shear band. When identical cutting conditions were studied for a similar material without strain-softening, no band formed. However, a curve of similar shape and location was found to exist in the workpiece by defining it as the contour of the smallest of the maximum shear stress values occurring at any point through the chip thickness. It was concluded that chip formation in both continuous and discontinuous chips is governed by the maximum shear stress level along this curve. Discontinuous chips form when the smallest maximum shear stress across the chip reaches the instability limit of the material. Otherwise, continuous chips are produced.

DIAMOND GRINDING OF OPTICAL SURFACES ON BRITTLE MATERIALS

By

Leonard E. Chaloux, Product Manager
Precision Machining Systems
Pneumo Precision, Inc.
Keen, New Hampshire 03431

Through our previous marketing efforts and discussions with various potential customers in mid-1984, it became apparent that there was a growing need for a grinding machine capable of producing aspheric optical surfaces on various brittle materials that could not be directly machined with single crystal diamond tools. Some of the materials of prime interest were ceramics, carbides, and optical glass. Furthermore, many of our customers wished to have the ability to use the same machine for single point diamond turning as well as diamond grinding. This would allow maximum utilization of the equipment for production of various types of optical components in a wide range of materials.

Upon further evaluation of the applications for diamond grinding, we determined that a major portion of the market would be served if we could diamond grind optical surfaces on parts up to 50mm diameter. Also, to fulfill the requirements for ultra precision lens molds, this grinder needed to be capable of producing accurate diameters and shoulders which were concentric and/or square to the aspheric optical surface. To satisfy the initial requirements set forth above, we concentrated our Research and Development efforts on the design of a production grinding attachment that could be added as an accessory to an existing diamond turning lathe.

This paper will focus on the information accumulation during the development grinding of tungston carbide and silicon carbide workpieces on a PPI Ultra-2000 Lathe utilizing this grinding attachment. The following key areas will be discussed in detail:

- I. Design of the Grinding Attachment
- II. Process Development
 - a. grinding wheel selection
 - b. wheel dressing
 - c. wheel setting
 - d. cutting parameters
- III. Restrictions/Limitations
- IV. Test Results
 - a. form accuracy
 - b. surface finish

There will also be a brief discussion regarding other areas of interest for future development in the precision grinding of optical surfaces.

APPLICATION OF MINIATURE BEARINGS TO MECHATRONICS

By

K. Kakuta
Nippon Seiko

REAL TIME CORRECTION OF SPINDLE ACCURACY

By

Thomas G. Bifano

And

Thomas A. Dow

Precision Engineering Laboratory
North Carolina State University
Raleigh, NC 27695

The widespread use of real time feedback control in precision engineering has been impeded by both the complex dynamics of the electromechanical structures and by the unavailability of practical control systems that could provide robust performance and relative economy of implementation. However, future industrial demands for precision parts will dictate active error compensation is required to supplement open loop machine control.

The task of accurately positioning a structure to microinch tolerances in real time presents a challenging area of application for automatic control. To combine sensors and actuators operating at their limits of speed and resolution into a workable error correction scheme necessitates innovative development of practical control techniques. This paper documents the development of a high speed real time feedback control scheme for improving the accuracy of a ball bearing spindle. Through this specific example, some general guidelines for the use of high speed, real time control of position will be introduced. Additionally, the benefits of active error correction will be demonstrated, and two practical control strategies will be compared.

Common to all real time control systems are three distinct subsystems: a measurement sensor, a control algorithm (implemented either in software on a digital computer, or in hardware on a dedicated circuit), and an output transducer. In the specific case of a system controlling the linear position of a structure in real time, a displacement measuring gauge is the sensor and a linear actuator is the output transducer. Typically, the control algorithm for such a system is tailored to the characteristics of both the sensor and the actuator, and to the dynamics, (e.g. linearity, natural frequency, stiffness, damping, etc.) of the structure being controlled.

The design process includes the selection of appropriate sensors and actuators along with modelling of the structure to be controlled. A straightforward integral controller has been presented, along with some methods for improving the response of this control strategy by the incorporation of added damping in the actuation loop. The results of this integral control study paved the way for a novel direction sensitive control system which greatly simplifies the complexity of both the sensor and the control algorithm, without sacrificing robustness or time response.

Although these results are presented for a specific control problem, the design process is similar for many positioning problems in precision engineering. Additionally, the algorithms (most notably the direction

sensitive controller) are largely independent of the specific structure being controlled. Versatility of application is designed into the algorithms, which rely on straight forward error compensation and direct, simple implementation.



Precision Engineering Laboratory

**North Carolina State University
Raleigh, NC, 27695-7910**

(919) 737-3024

